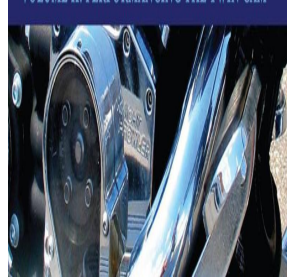
DONNY PETERSEN

"THE WORLD'S MOST-READ HARLEY TECHNICAL JOURNALIST"

# DONNY'S UNAUTHORIZED TECHNICAL GUIDE

TO HARLEY DAVIDSON 1936 TO PRESENT

VOLUME II: PERFORMANCING THE TWIN CAM



# Donny's Unauthorized Technical Guide to Harley-Davidson, 1936-Present

## Volume II: Performancing the Twin Cam

Donny Petersen

iUniverse, Inc.

New York Bloomington

Donny's Unauthorized Technical Guide to

Harley Davidson 1936 to Present

Volume II: Performancing the Twin Cam

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E. & O. E.

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# Acknowledgements

I acknowledge my editor and publisher because they have been very much a pain in the ass.

Wow!

I thought writing The Encyclopedia of Harley-Davidson Technical Knowledge, 1936 -2008 would be easy. After all, I have been writing a minimum of one article per month since 1989; first for Canadian Biker Magazine and then for American Iron Magazine since 1992. I had the mistaken notion that I would simply sew a bunch of articles together and wind up with a work of art.

It is not that simple.

Take the title, for instance; the publisher informed me an encyclopedia started with the letter A and ended with the letter He pointed that out because I was starting with the letter as in Twin Cam. The compromise became The Technical Guide to Harley-Davidson, 1936-2008.

The next question centered on whether Harley-Davidson knew I was writing the book. I replied that the Motor Company did not, nor did anyone else on the aftermarket side. I said that I wanted to be able to write my thoughts without marketing pressure from any company. The compromise became The Unauthorized Technical Guide of Harley-Davidson, 1936-2008.

Then the publisher wanted to know why I thought I could

embark on such an ambitious venture without both of us falling flat on our faces. He asked for credentials and became impressed, especially with the fact that only the name Donny might appear on the cover of an international magazine. If the name Donny is instantly recognizable by readers and they buy the magazine because Donny is in it, that cachet should change the title one more time to Donny's Unauthorized Technical Guide to Harley-Davidson, 1936-2008.

He did not ask a very important question.

Will the technical guide end in 2008? Of course, it will not. It will continue until the demise of current models and then the new models will evolve into separate volumes of their own. The title should end with not 1936 to Oh!

Once issued with an ISBN number, the book title cannot change. The ISBN is the way books are identified by

publishers, bookstores, libraries, and the like.

So, guess what. Each year of current model books will have corrections, additions, and a new chapter on the coming year.

In all twelve volumes only two models are currently available, the Twin Cam and the Sportster. The volumes on non-current models, like the Evolution or the Shovelhead, will stay pretty much the same since not much new information will be forthcoming. So, in the winter of 2008, an updated 2009 version of Volume I: The Twin Cam will arrivewith an altered main title which will now read Donny's Unauthorized Technical Guide to Harley-Davidson, 1936-Present. It will also have a new ISBN number because the word present replaces

In addition, I will attempt to correct the typos, as well as

some misinformation, notably the definitions of Octane and Detonation. These definitions will also be correct in this book, Volume II: Performancing the Twin

Confessions

All of the following mistakes are corrected in the updated in Donny's Unauthorized Technical Guide to Harley-Davidson, 1936-Present: Volume I: The Twin

In the first printing of Volume I: The Twin Cam, I wrote, obviously in a state of sleep deprivation--how else could I have made such a stupid tightening a chain or belt is done at the loosest spot. Well, every mechanic knows, or should know, that a chain should adjust at the tightest spot! If adjusting at the loosest spot, the tightest spot may snap or cause rapid adverse wear. I thought of turning my foolishness into a contest about what my dumbest mistake was, the prize being

of all twelve volumes, but I needed this confessional.

Furthermore the section, An Overview of the Twin Cam Fathead repeats itself twice early on in the book. Fortunately, it is not very long.

I now proof on printed pages because for some reason typos and dumb mistakes get lost on a computer screen.

I do not think I have ever easily learned a lesson in my life. Learning the hard way has one redeeming feature. I rarely make the same mistake twice, once educated, because it indelibly imprints on my psyche.

I acknowledge the patient people that made me do everything right in the ways of the publishing world because, after all, if books do not have a common format of grammar

and all the other stuff, then there would be no uniformity. The readers should have some guarantee that books will follow a set of rules that provide consistency. No one ever notices this. However, if differing formats were commonplace, readers would notice constantly changing and disconcerting spellings, grammar, positioning of text, and all the rest.

I acknowledge my former girlfriend Sandra whose leaving now allows me to write in peace. I wish her good luck on finding all she wants.

The Swat Team Raid

I acknowledge the Canadian legal system that I once thought of as going easy on those charged with various offences.

I learn the hard way once again; the system is anything but

that. From the predawn raid, with kicked-in doors, concussion grenades, and two swat teams; lying naked on the floor handcuffed, disoriented with a couple of cops wondering aloud why I hadn't defecated myself from the trauma.

I kiss the ground in gratitude that Sandra was not there, for she would have risen in indignation, and been knocked down, which would have lead to my defending her with me receiving resultant rifle butts to the head and repeated electric stun gun shock tasering.

I have no children, but the raid would have traumatized kids for years to come.

For the first time in my life, I am so happy that I did not have a large pet dog. Who knows what would have happened to him?

They cannot traumatize me if I am alone. What happens to loved ones certainly would affect me in a big way.

Neighbors lining the street assume I must have done something terribly wrong to warrant the attention of an estimated seventy-five police.

Very cold jail cells prevent sleep.

Newspaper accounts sensationalize with your face front and center.

The court appearances, the sheer expense, restrictive bail conditions..., and the waiting all combine to punish before a finding of guilt or innocence.

I am lucky because I am out on bail; there are others sit in jail perhaps for over two years awaiting trial while their marriages disintegrate, pensions are lost, businesses fail, houses and possessions lost, and on it goes. The innocent have nothing once released, their lives destroyed. I now know why recidivism rates are so high in our society. How can one survive after a lengthy jail sentence without proper clothing, money, a job and emotional support?

The damage to your house is unnecessary when a phone call or knock on the door would accomplish an arrest.

Disclosure of evidence that leads to the arrest takes a year.

I knew I had done nothing wrong. Their evidence indicates that. Here I sit, waiting for a judge to rule. How long…I do not know but it may be years.

I am thankful I have friends and the money for a competent lawyer.

Now I understand why innocent people are in jail. No money equals a lousy defense in an adversarial system. A legal mind needs to pore through a pile of disclosure evidentiary paper twenty feet high.

Therefore, even innocent as I am, the punishment is still severe. On going to jail if found guilty, it is only the second last punishment of many. The last punishment is the assumption by authorities and society that you must be guilty if charged. This stigma remains forever. The assumption becomes a technicality that prevents conviction, not innocence.

The bright side is that I am able to do some writing.

In the end, this experience will have great value in again pointing me in the right direction in life from such a terrible experience.

For now, I have something that few people have. I have the knowledge of who are true and faithful friends. I also have knowledge of the bastards that pretended to be what they are not.

This alone will make the total experience worthwhile.

My case has a publication ban so I will not be able to write more until this is over.

Rest assured more information is forthcoming.

Besides how can one have an interesting life if not been kicked around once in awhile?

Now, some kind words from my old pal Lenny Creed, whose magazine, Canadian Biler Magazine gave me my first break.

25 Years with Donny and Len

When Donny asked me to write a piece, and said he would put me in his second volume, I thought how ironic as it was that 25 years ago today I was back in Toronto doing a bike show and I asked him to do the same thing!

I remember that day as if it was yesterday. When I walked into Heavy Duty Cycles in Toronto, Ontario and asked this burly biker behind the counter “is Donny Petersen here” the

reply was “depends who wants to know!” Well, being a smart guy myself I introduced myself as Len Creed publisher of a motorcycle magazine called “Canadian Biker” and I’d like to talk to him about doing something in the magazine (I was thinking advertising but seems I had to get past this big guy first, or at least talk to a manager; someone a little more friendly!).

Then a big smile came out of this good-looking brute and said, “I’m Donny and I’ve heard of your magazine. Interesting, but I don’t need any advertising if that’s what you’re thinking.”

Well, I shook his hand and went into my song and dance anyways since this was to be a couple of busy days pounding on doors trying to find sales and input from the East. The more we talked, the more the wheels were turning with the

idea of a feature bike or something on the shop. I had gotten something interesting but no advert yet. When our time ended, I walked away wondering if this Donny guy would ever follow through with a promise to send in a feature to me. Only days later I got a call from Donny saying he checked me out and a feature bike were in the mail to me (yes, it was mail or courier in those days and a longhand written article with a few pictures). Man, do I remember trying to decipher everything correctly from the handwriting besides doing the feature justice so we could get more of the same in the future.

That was a start of a friendship that continues today and the celebration of a new career for Donny getting published. I remember him telling me it made his day, but more importantly his mother's day, because she always wanted him to be a teacher/writer.

Little did we know way back then that this would all come to fruition; a couple of bikers coming together with some

ideas and making our dreams a reality.

Donny wrote a regular column for me, over those next five years, titled “Department of Acceleration” and every year we got together at bike shows, events, and rides to update each other. My biggest accomplishment with Donny (other than writing for the magazine) was to get him to use a computer. It was always “old school” for Don and he just could not do it on what I was paying him (or so he thought). It took about three years to get Donny to even to look at a computer, let alone learn to type again from high school--of which Don had passed with honors. It took the help of another friend of ours, Marty MacDougall, who was at that time doing the photo-shoots for the feature bikes and Marty had all the latest tech stuff. Between the two of us, we ganged up on Donny and showed him the way to even more success. He started writing for more magazines that paid better, started a teaching job at a local college, and discovered the Internet. He built a website

that included many of his writings, and columns over the years followed by a host spot on Biker TV in the tech segment.

Donny is now one of the most read technical Harley writers in today's market and I cannot tell you how proud I am of him and his accomplishments (he will probably want me to pay him for those first few years!). He is just a life long biker I like to call a friend and look forward to our next ride together because you never know what the two of us will come up with next!

Len Creed, founder of Canadian Biker Magazine and Canadianbiker.com and now Mybikespace.ca as I head into retirement just ahead of Donny "The Man" Petersen!

# Preface

First, I am going to include a full listing of acronyms, the reduction-letters that replace unwieldy long forms in each volume, beginning with this one. I will retrofit the acronym listings to Volume I: The Twin Cam in the 2009 reprint update.

What do those damn letters stand for?

Acronyms can be very confusing to those that do not immediately recognize what the letters mean. They also prevent learning and an enjoyable read.

For example, if presented with EITMS, how is one

supposed to know that it is the acronym for Engine Idle Temperature Management System, a very necessary cool-down feature in modern EFI (electronic fuel injection)?

New in this Volume II: Performancing the Twin Cam and continuing in the next ten volumes will be entertaining but educational vignettes' called Donny's Blasts from the

Donny's Blasts from the Past

These anecdotes will tell it the way it was. No refined and pasteurized corporate images existed back then. They were real characters, which, in part, passed on to Harley-Davidson the rich heritage the Motor Company enjoys today.

The H-D fashion of the day consisted of worn and greasy blue jeans, heavy scuffed boots, plaid shirts, and sun-faded WWII bomber jackets; dressing up meant putting on a black

long-sleeved shirt.

The vignettes are not always complimentary but they are as true as memory will serve. Of course, storytellers add their own twists.

## Mouldy and the Land Speed Record

I will revisit Mouldy, the irascible local dealer who attempted to badger Fast Eddie, his head mechanic, into attempting a land speed record on a Knucklehead in the 1940s.

The wind was too strong, causing one hapless rider on Eddie's Knuckle to lose a leg.

Pissed, Eddie hopped on a Snortin' Norton and set the record, not much good for a Harley dealership to crow about their products.

It was a man's world, where the dealership personalities--Mouldy, Fast Eddie, Shy Ron, and sometimes Fats--would hop on their rigid frame Knucks, Pans, and the odd Flathead 80 after Saturday closing to ride five-hundred miles and camp out before returning to work Monday morning.

Shovelhead Warranty Blues

In another blast from the past, Donny will tell about Teddy, the circa-1980 service manager at another H-D dealership, frustrated with the volume of Shovelhead warranty claims.

Teddy set up a betting pool for the mechanics and himself.

They would all gather round with each Shovelhead delivery and bet if the new Harley would make it out of the parking lot before breaking down.

Readers should not interpret Donny's Blasts from the Past as put-downs or other types of negativity but as glimpses at the way, it was. Those days were rough and ready, and they were fun. Political correctness did not exist on any level.

The era's bikers were sexist, as was society in general. Donny's Blasts from the Past may contain sexist commentary. However, this commentary's design is to suggest the flavor of days past.

I may use some words such as Limey which I guess might interpret by some as racism directed against the English. There is no intent as such. A Limey bike simply meant a BSA,

Triumph, Matchless, or Norton motorcycle to distinguish them from the only other two games in town, Harley and Indian. Riders, back in the day very much respected and loved the Limeys.

Donny's Blasts from the Past will serve to demonstrate just how far Harley-Davidson has evolved. They will show just how good the Twin Cam is relative to what has gone before.

Harley gained popularity because of the personality quirks of both the bikes and the people around them. In a homogeneous world with each big-box store identical to the others, present and future vignettes will fall by the wayside.

Donny's Unauthorized Technical Guide to Harley-Davidson, 1936 to Present will comprise of the twelve volumes listed below. A brief synopsis follows for the reader's perusal.

Twin Cam

the Twin Cam

Evolution: 1984 to 1999

the Evolution

Sportster: 1952 to Present

It All Works

1966 to 1984

H-D Technical Encyclopedia

Knuckleheads, and Flatheads: 1929 to 1973

Tips

the Dream

Each volume will describe, discuss, and analyze a different technical subject within the world of Harley-Davidson except Living the which will be about my riding exotic locales and the Rode-Mode survival tips I learned the hard way.

Volume I: The Twin Cam lays the groundwork for all that follows in Donny's Unauthorized Technical Guide to Harley-

Davidson, 1936 to Present. I view the Twin Cam as a transitional model into the future as the Factory must deal with ever-increasing EPA dictates especially in 2010 and beyond. Let us not forget the influences of the California Air Resources Board (CARB), which many times may predate EPA guidelines.

I think the V-Rod is the Twin Cam's design future.

Volume II: Performancing the Twin Cam is a dynamic introduction to the quest for speed.

The Twin Cam is the best platform for performancing in the world of Harley.

Carburetors, fuel injection, cams, headwork, strokers, big-bore, supercharging, turbocharging, nitrous oxide, and much

more are all hot topics for discussion.

A comprehensive chapter will also discuss the problematic CVO 110 engine.

I will redesign this engine, giving it the greatness it aspires to.

A chapter on vibration will discuss the inherent dangers beyond rider discomfort to support bearings as well as the rest of the motorcycle from performance and longevity perspectives.

Vibration and performance do not mix. I will talk about how to eliminate many types of vibration and how to minimize the rest through truing flywheel assemblies to balancing techniques.

Then I will take the reader through building a very reliable 124-cubic-inch engine using stock Twin Cam engine cases, a feat no previous H-D engine has managed to perform.

Volume III: The Evolution: 1984 to 1999 is the bike that many credit with saving Harley-Davidson from bankruptcy. Why did it become the chosen one? After all, there was a more progressive and practical bike in the works. However, Harley-Davidson is not about practicality.

How did H-D become the darling of American capitalism?

I will explore the Evolution from different perspectives, explaining why it is so much faster and more reliable than its Shovelhead predecessor is. However, there is a lot wrong with the Evo that loyal riders love to overlook. I will get my hands dirty demonstrating the many foibles of the Evo.

Volume IV: Performancing the Evolution will describe the hop-ups available for the fastest of the Harley Big Twins up until 1999. It is easy to make the Evo faster within the limitations of the engine and retain reliability. I will explain how and show you what to do, and perhaps more importantly, what not to do.

Volume V: The Sportster: 1952 to Present will begin with Flathead K models and lead into the birth of the modern Milwaukee Vibrator, the 1957 overhead-valve Sportster. The Sportster lost its vibrator status in 2004, entering a new era of touring capability. I will chronicle the Sportsters many mechanical changes over the years.

The historically ignorant and sexist derisively relegate the Sportster to girl's bike status. Originally, a man's bike, the Sportster dominated racing as well as the hardcore outlaw

lifestyle that all modern riding clubs like HOG are based on. Consequently, it is the most customized Harley of all time.

Performancing this ground-pounder will be a hot section of the book.

I will then explain the future and all the changes that will occur to meet the 2010 and later Environmental Protection Agency (EPA) emissions restraints. There will be big changes in the Sportster to meet compliance that began with electronic fuel injection for all models in 2007.

Surely, the Sportster also faces a major redesign in the very near future. Perhaps the Buell 1125R is the harbinger of change. Never mind perhaps! It will be the future of the Sportster.

Volume VI: How It All Works, will be a techie gearhead's dream. I will explain the component parts of the engine and subsequent Powertrain from valve springs to sprockets, examining their individual functions and interrelationships.

A total understanding of the many V-Twin's, the Flathead, Knuck, Pan, Shovel, Evo, Twin Cam, K-model, and the Ironhead plus Evo Sportster engines in addition to the beginnings of the V-Rod will evolve in the reader's mind. With this understanding, analysis will follow and thus the ability to diagnose.

Volume VII: Shovelheads: 1966 to 1984 will discuss their development from the Panhead. The Shovel lost our affection with the advent of its more reliable successor, the Evolution. The Shovelhead is back with a vengeance feeding the "bobber" boom the custom industry is currently enjoying. The Shovel has become one cool bike.

Volume VIII: Lubrication is the all-time favorite subject of those that discuss their bikes and mechanics.

Which brand oil is the best? The half-truths, marketing claims, and mythology surrounding oil form a maze that few penetrate. I am going to remove the hype and give the real deal. Which oil will win? The combatants will be out of the ground fossil oil versus fabricated synthetics.

Myth will separate from fact, perhaps giving the reader his first unbiased look at this intriguing subject.

I will then go in depth about Harley's marking their spot. Why do traditional oil leaks occur? How do we fix them? Why does your Harley puke oil out the breather? I will explain what is wrong and why and then explain the fixes.

Volume IX: The H-D Technical Encyclopedia will be a calculation reference bible.

There will be a comprehensive section on engine formulae that will cover calculations of every aspect of Harley-Davidson mechanics and performance. This will follow with explanations on how to make them work for you. The book will discuss such topics as taking everyday practical engine formulae like calculating compression ratios, explaining the math, and then applying the results to bump an engine's compression.

I will also discuss a myriad of interesting topics like calculating cubic-inch displacements, transmission gear ratios, and final drive ratios.

The reader will learn how--just like the experts--to modify

his particular bike's specifications, altering them to tailor his specific needs and wants.

Volume X: Panheads, Knuckleheads, and Flatheads: 1929 to 1973 will chronicle the development of the Flatheads from 1929 to the demise of the pie-wagon trike in 1973.

Yeah, I know, I screwed up the title of the book. It should read 1929 to Present not 1936 to My knuckle brain was thinking of the Knucklehead, which did begin in 1936. Some really interesting stuff happened between 1929 and 1936 and I will cover it all.

Next, I'll discuss the first hemi head Harley, the legendary cast-iron Knucklehead built from 1936 to 1947.

This intriguing volume will round out with an exploration

of the beloved 1948-65 Panhead.

To quote the sexist vernacular of the day, “It takes a man to ride a Pan.”

Volume XI: Tech Tips will offer hundreds of practical tips on Harley-Davidson mechanics, short cuts, roadside repairs, and general information that will be intriguing for gear heads and casual readers alike.

One of yesteryears tips concerns a roadside fix using Comet cleanser to prevent engines from smoking oil out the exhaust. Blowing this fine abrasive into the carburetor throat reseats worn-out piston rings. This treatment stops smoke emanating from the exhaust pipes and, more importantly, prevents the spark plug from oil fouling, leaving the rider at the side of the road.

This volume will read like Confucius's sayings.

Volume XII: Living the Dream will contain roadside technical advice interspersed with my biking adventures. Sprinkled throughout will be Rode-Mode practical guidance that every high-miler learns the hard way, such as surviving impacts at speed with dogs as well as bears, cows, deer, moose, and a flock of Canadian geese. My riding friends and I have encountered them all.

Living the Dream will chronicle my rides around the world, from crisscrossing Europe and North America, to riding Central America and Brazil.

The riveting circular ride around South Africa to God's Window is in a most beautiful and friendly place.

However, there are ten different ways to die each day.

# Introduction

These twelve volumes comprising Donny's Unauthorized Technical Guide to Harley-Davidson, 1936 to Present is a culmination of a life's work that is not yet complete.

Firstly, the volumes that discuss Harley-Davidson's that are still in production will not end with the year 2008 or 2009 or 2010. These volumes will update every year. There will be a banner across the covers to denote 2010, 2011 and on it goes until Harley-Davidson ceases production of a particular model.

Therefore Volume I: The Twin Cam will reflect updates beginning in 2009. I will update Volume II: Performancing the

Twin Cam as necessary. The year of updates will be on a banner on the cover. Volume III: The Sportster 1952 to Present will begin updated versions in 2010. Knowing me the way I do, I can see myself adding to all volumes every year as I am on a constant learning curve that never seems to end.

I realize constantly how little I really know relative to how much I still have to learn.

When the Motor Company replaces the Twin Cam or the Sportster with new models, there will be a thirteenth and fourteenth volume for these new Harley-Davidson's. I also plan a future additional volume on the V-Rod as I think Harley-Davidson will incorporate its design or major parts of its design into the motorcycle that will replace the Twin Cam.

I view the Twin Cam as a transitional model into the

future as the Factory must deal with ever-increasing EPA dictates especially in 2010 and beyond. Let us not forget the influences of the California Air Resources Board (CARB), which many times may predate EPA guidelines.

Surely, the Sportster also faces a major redesign in the very near future. Perhaps the Buell 1125R is the harbinger of change. Never mind perhaps! It is the future of the Sportster.

It follows that I think the V-Rod is the Twin Cam's design future.

# Chapter I

## The Platform for Performance

### Mechanically, the Best Engine Since 1903

Much of the Twin Cam engine is a bulletproof platform for the future of Harley-Davidson with this genre engine. Designed with durability and reliability in mind, the engineers were expecting a minimum of one hundred thousand miles before a rebuild.

Excepting the 1999-2006 problematic timing chest components, which the performance mechanic easily fixes, the longevity expectations are likely understated.

Many times, my writing necessarily focuses on the problems and fixes associated with Harley-Davidson motorcycles. This may create negativity after awhile, simply because the reader may lose sight of the wonderful aspects of the Twin Cam engine relative to what has gone before.

## Was the Evo a Great Engine?

Sure, the 1984-1999 V2 Evolution was a great engine, relatively speaking. Perhaps it was not so great when one becomes familiar with the various foibles over a highly successful sixteen years of use.

The Harley rider is still very forgiving of any problems. Many seem to suffer memory lapses over warranty issues like engine casing porosity and cracking, and let us not forget

those forever-leaking base gaskets. I attribute these wonderful memories of Evolutions to the positive aspects of the engine and comparable bikes of the time. Many of these riders were fresh off their beloved Shovelheads.

The Evo was very much faster than what went before because of the D-shaped combustion chambers, the introduction of the squish band and flattop pistons. For the first time, the air-fuel mixed better, making the Evo so much faster than the inefficient hemi head 1966-1984 Shovelheads.

Evo reliability also became a legend in its own time, as the V2's reputation was very deserving relative to the Shovel.

If the reader thinks I am kidding, let me relate a Donny's Blasts from the Past true story circa 1980. A large local multi-line dealership could only laugh in embarrassment at the breakdown rate of the Shovelhead when comparing to their

Japanese products even though these bikes outsold H-D by about ten to one.

Donny and H-D Design Engineer with Prototype Twin Cam

I am standing with Bob “Koby” Kobylarz, the chief engineer of the powertrain division of Harley-Davidson Motor Company behind a prototype Twin Cam engine in March 1998. The revealing of this engine by Harley-Davidson born from the

ultra-secret P22 project to Buzz Kanter, Chris Maida, publisher and editor respectively of American Iron Magazine and myself was one of the most exciting moments of my mechanical We signed secrecy agreements with the Motor Company not to reveal what we had seen until half a year later. Koby and I are standing just outside the Harley-Davidson Archives in the old six-story building at 3700 Juneau Avenue in Milwaukee Wisconsin.

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Betting on Shovelhead Warranty Claim Volume

My buddy Ted, the service manager, had four stacks of warranty files on his desk at any given time. The Honda, Suzuki, and Yamaha piles were each about ⅜" thick. The Harley-Davidson Shovelhead pile was always 6" thick and

some days so high it was in danger of falling over.

Teddy and the mechanics had a pool set up to bet on how far the newly delivered Shovels would go before the writing of the first warranty claim. Many bikes did not make it out of the parking lot before Ted, in resignation, wrote the first claim.

Evo reliability was astonishing in light of the only comparisons available like its predecessor the Shovelhead.

I am a mechanic that has traversed the Shovelhead, Evolution, and Twin Cam years. The increases in reliability of the Evo over the Shovel and now the Twin Cam over the Evo are astounding.

Harley-Davidson has made huge strides in this area. Some individual components in the Twin Cam engine will last many

multiples of the magic one hundred thousand mile figures. This is a telling statement in an age of planned obsolescence when vehicles and parts have a built-in life expectancy.

## Bulletproof Parts

The 1999-2002 engine casings that utilize the opposed and tapered Timken bearing sets to support the sprocket shaft in the left casing successfully since 1956 are a superlative design.

In my opinion, these 360 specially formulated proprietary alloy casings, with care, will be serviceable for a half-million miles and beyond. For further testament in Chapter XII: Cubic: Strokers and Big-bore I will build a very reliable behemoth 124-cubic-inch Twin Cam using stock Twin Cam engine casings!

The 2003 and later casings will not be as fortunate since the engineers have eliminated the very reliable sprocket shaft Timken bearing sets in favor of a single roller bearing. The year 2000 and later Softail engine right-side pinion shaft roller bearing has supplanted the Timken bearings on all 2003 and later Twin Cam engines.

I do not feel this type of bearing that works wonderfully with the up-down stresses imposed in the right casing will be able to withstand the extra rigors resulting from the throbbing sideway power pulses of the transferring power stroke down the powertrain to where the rubber contacts the asphalt. The Timken bearing sets are more tolerant of crankshaft flexing at high rpm. Interestingly, both current manufacturers of aftermarket Alpha Twin Cam casings, S&S Cycles (www.sscycle.com), and JIMS USA (www.jimsusa.com), use the Timken bearing setup instead of the roller bearing on the sprocket side for all year applications.

JIMS USA also makes Beta Softail Twin Cam cases that use the sprocket shaft Timken bearing sets for all years also.

However, if using 2003 and later stock engine cases for a performance application that uses a roller bearing on the sprocket shaft does not bother you, I will describe the various ways to overcome this potentially bad performancing situation again in Volume XII: Cubic Inches: Strokers and Big-bore.

Twin Cam engine case manufacturing only requires clamping five times for complete machining. Clamping distorts the alloy, which can cause machining distortions as the alloy bounces back to its former shape before clamping. Less clamping means a superior product. In comparison, the Evo cases require clamping thirty-two times.

The permanent mold cylinder heads using 242-alloys is also outstanding in their quality construction from a longevity point of view.

An efficient Twin Cam head design using a bathtub combustion chamber on the inside accompanies a 60% increase in finning on the outside for superior heat exchange into the atmosphere. I feel these heads will be around many decades from now, rebuildable for following generations of riders wanting original stock parts.

However, the respective heads do not flow enough air for a stock TC88 1450cc or for the TC96 1584cc engine. In fact, they flow less air than a set of smaller 1340cc Evolution heads. While the TC96 heads have improved flow over the TC88 heads, it is still not enough for the performance-minded rider. This airflow concern also need not bother the reader, as

I will elaborate on this issue later in this chapter and will remove the impediments to airflow in Chapter VIII: Eliminating the Performance Obstruction.

The barrels, also known as are shorter due to the smaller stroke of 4.000” from 1999-2006 and 4 ⅜” in 2007 and later Twin Cams and combine with a higher deck height on the engine crankcases. The deck height is where the barrels mount onto the engine cases.

Shovelheads from 1978-84 and Evos from 1984-99 used longer cylinders because of an elongated 4 ¼” stroke and lower cylinder decks on the crankcases. The Twin Cam cylinders are fatter due to the massive 3 ¾” bore and the increased fin area. The cylinder sleeves that house the pistons are thick (.247” compared to the Evo’s .173”). This ¼”-thick sleeve liner provides stability for the piston as it changes thrust axis particularly at the bottom of the stroke.

The theme of rigidity in this platform engine comes into play to provide stability for component parts. This is due in part to the thickness of the liner but also because of the new spiny lock iron sleeve cast into the high-pressure, die-cast cylinder.

Mounting the engine and transmission together is another superlative move on the part of the Motor Company. This provides much rigidity to withstand the torques twisting forces traveling down the powertrain. No longer does the primary system need to function as a support, although it still lends its unifying force.

Let us get back to the cylinders and their mating black Teflon-coated hypereutectic aluminum pistons. Oil-cooling jets relieve about F from the piston crowns. Hypereutectic is a

cast piston with greater than 12% silicon content. The silicon crystallization prevents heat transfer to the more vulnerable piston skirt thrust areas, allowing them to run cooler. They are the strongest of any pistons previously used by the Motor Company, are most resistant to piston-to-wall scuffing, and least likely to seize up in spite of the very close fitment to cylinder wall.

I broke in both my 2006 TC88 FLHXI and my 2007 TC96 FLHXI at 55-75 mph and 60-80 mph respectively in cold weather. I disassembled and performanced both bikes soon after the break-in period. The piston ceramic coatings had nary a mark on them. When Harley-Davidson does it right, H-D does not fool around!

The combination of pistons and cylinders with the possibility of five over-bores will ensure longevity for over five hundred thousand miles or more, maybe even double if religiously cared for.

Piston wristpins have always been a durable engine part. The Twin Cam wristpin is big, and I mean big, with a correspondingly large wristpin bushing in the matching connecting rod that is meaty enough for the biggest engine. An Evo wristpin is .792" thick while the Twin Cam's is a whopping .927" in diameter.

The right case support bearing for the pinion side of the flywheel assembly is a huge roller bearing about 3 ½" in diameter, about double that of the relatively skinny needle roller bearing in the Evo that sits on a beefy pinion shaft forged as part of the right-side flywheel.

The Twin Cam pinion shaft is .400" thicker than its counterpart Evo at the pinion bearing location is.

The three-oiling-hole crankpin is muscular and .420” thicker than the diameter of the now skinny single-oiling-hole pin used in the Evo.

All engine covers like the nosecone or timing chest, the split two-piece rockers wafers, and the guide block covers are just that…covers and will exceed one hundred thousand-plus miles of leak-free usage. No more will these covers act as load-bearing supports for other engine parts as they have in all engines before 1999.

The connecting rods are the biggest and strongest ever used by the Factory. In fact, they are bigger than aftermarket con rod sets designed for the big inchers.

The theme of size continues with the captured three sets of eighteen needle rollers in copper-coated cages, which help

break-in procedures, provide durability, and offer easy-wear spot identification. Captured is an engineering term to describe the type of bearing cage, which supports each of the eighteen bearings per cage around all sides.

The cylindrical roller bearings are bigger than the Evo's or the Shovel's. There are three more bottom end bearings than in the Evos and the late Shovels. They are the same diameter but longer in length.

I could go on, but the reader now understands what a platform for power this engine base really is.

In my opinion, it is unusual for a modern manufacturer to make parts as durable, with as much built-in longevity as Harley-Davidson has with both the Twin Cam TC88 and the TC96.

The ongoing theme of rigidity ensures that the Twin Cam will not only be the best- handling Big Twin at speed and in adverse handling situations but will transfer horsepower through the powertrain without the torsional twisting so common with previous Harley-Davidson's.

For extensive information on the technical aspects of the stock Twin Cam TC88 engine see Volume I: The Twin Chapter I: March 1998: Introduction of the Twin Cam and for the TC96 engine see Chapter VIII: The 2007 TC96 Engine also in Volume I: The Twin

Stress versus Parts Longevity

Individual simple parts from the 1999-2002 huge rubber-mount pinion ball bearing to more complex pieces like the head castings and the superb engine casings do not appear to have built-in obsolescence.

Properly trued flywheel assemblies, at the Factory to one-thousandths of an inch (.001”), or less will surpass the 100,000-mile marker easily. Those that are out-of-whack anywhere from three or four-thousandths of an inch and up will damage mating components like the oil pump and the cam support plate, eliminating the particular engines they are fitted in from the vaunted high-miler club.

Other weaker parts, such as the 1999-2006 timing chest components and 2003 and later sprocket shaft bearing, will be hard pressed to keep up with the high-miler parts, some of which I have already listed.

The Softail 2000 and later pinion bearing is still very big but smaller than the 1999-2002 rubber-mount pinion-side ball bearing.

The jury is still out on the timing chest changes instituted with the 2006 Dyna and carried over into all 2007 and later Big Twins. The changes for the most part look positive, but time and miles will be the evaluator.

Engine Stress

Why would the same pinion bearing be so reliable in this situation but not so durable in the other, as the sprocket shaft bearing? After all, this bearing supports the flywheels at either end on their shafts as they pass through the engine casings into other compartments.

Stress is the short answer.

On the pinion shaft, the ball bearing simply and effectively allows the shaft to rotate in its center. The only stressors are

perpendicular to the rollers.

On the sprocket shaft, also known as the power shaft, pulsating power surges thrust out, driving the primary sprockets and chain, turning the transmission while the clutch engages, and, finally, powering the final drive, turning the wheel.

Different stresses require different types of bearings.

Comparing Twin Cam Flywheel Support Bearings

Pictured from top to bottom is an H-D 9028 tapered Timken sprocket-shaft-bearing kit in use from 1999 to 2002 in both Alpha Dyna and touring models and Beta Softail models to support the flywheel sprocket shaft in the left-side engine casing. The next single roller bearing is H-D 24623-99, H-D 24623-99A, and H-D 24623-99B pinion bearing used from 1999 to 2002 on Alpha models only to support the pinion shaft in the right-side engine casing. The third bearing is the same early pinion bearing but did not self-destruct, which would be highly unusual because of its sheer size and strength married with proper function. I destroyed it to show the reader the component parts. Lastly, on the far right, H-D 24604-00, H-D 24604-00A, H-D 24604-00B, H-D 24604-00C, and H-D 24604-00D sprocket shaft bearing and pinion shaft bearing in use on all B models from 2000 to 2006. This roller bearing is also adapted for use on all Twin Cam A models from 2003 to 2006 on both pinion and sprocket shaft sides of

the flywheels. The 9028 Timken inward-facing tapered bearing kit is comprised of two tapered races fitted with two tapered roller bearings and separated by a precise spacer-shim that determines flywheel endplay inside the engine cases. There sixteen different-sized spacer-shims vary in size from .0895” to.1195” in successive gradients of 20-thousandths of an inch. Different sizes enable a mechanic to choose his own end spacing. For example, a stroker will require more spacing as more heat generates, possibly causing parts to expand more. After installation of the bearings and races, a thicker race may be required to increase endplay, or a thinner one may be necessary to decrease flywheel side movement inside the engine casings. The races press into the sprocket shaft hole in the center of the left-side engine casing with the tapers facing each other. The inside tapered Timken roller bearing presses onto the sprocket shaft with the taper facing the engine casing. The flywheel set installs into the casing. The spacer fits loosely on the sprocket shaft and installs from the outside of the casing. Then the other tapered roller bearing

presses onto the sprocket shaft and into the casing from the outside. The spacer separates the two bearings, determining endplay for the flywheel set. This bearing kit in slightly different variations has been in successful use from 1956 to 2002. The discontinuation of the tapered Timken bearing kits is a huge disappointment to the performance mechanic. These bearing kits never presented any problems and proved to be very reliable and durable. In my opinion, the H-D 24604-00 roller bearings that replaced the tapered Timken kits on the A models in 2003 and the B models from their inception in 2000 are well suited to withstand up-and-down forces. This is why they work so well on the right pinion side of the engine, where there is no side thrust forces. However, on the left sprocket power side of the engine there is powerful and pulsating side thrusting outward forces. I feel this type of roller bearing is ill equipped to handle this kind of situation over a long time-period, especially in performanced engines. I think the reason the bearing is so large is to compensate for these side-thrusting forces. So, why did the Factory change to

these roller-type bearings? Ease of installation, less parts involved and less expense come to mind.

Wrong Bearing Use

In the first year of the Twin Cam, two identical ball bearings were in use to support the outboard cam end journals in the cam support plate. The rear one self-destructed while the front one, mere inches away, rolls on effortlessly. Different stresses allowed one cam bearing to enjoy the ride while the other disintegrated.

Simply replacing the ball bearing as the rear cam support in late 2000 with a roller bearing solved the problem.

In my opinion, the rear-cam-journal support, Ina bearings used from 1999 to 2006, excepting the 2006 Dyna, are another inappropriately used bearing. We see damage to them

all the time at Heavy Duty Cycles. We always use B-148 Torrington bearings to replace them. Both Crane and S&S recommend this also. They are both needle-type caged roller bearings, but the Inas are smaller with supports to assist them in rolling square and true. The Torrington rollers are larger with more rolling area. In the 2006 Dynas and all 2007 and later Twin Cams, H-D still uses Ina bearings but has increased their inner diameters .125", from .875" to a full 1.000". I am sure these will be more durable than what went before, but I highly recommend replacing them with the equivalent B-168 Torrington bearings.

However, H-D has used the same type of Ina bearings throughout its five-speed transmissions since 1980. In this situation, they work wonderfully and very rarely need replacement even on transmissions twenty-nine years old! Of course, we replace them during rebuilds on general principles, but they are extremely reliable and durable in the

transmissions.

Different stresses for different bearings explain all.

## Twin Cam Radical Design Change

For the most part, Harley-Davidson has obviously overengineered many components in the Twin Cam engine.

The 88 cubic inch (1450cc) Twin Cam is the most radical Big Twin design evolution to occur in six decades since the historic inception of the first complete overhead valve Harley engine, the 1936 EL 61-cubic-inch (1000cc) Knucklehead.

The Evolution V2 (1984-99), engine signified the birth of the modern Harley-Davidson superseding the dinosaur technology of the last hemi, the Shovelhead (1966-84).

The Evo moved away from inefficient hemispherical combustion chambers and matching round-dome pistons to a modern squish combustion chamber with a mating flattop piston.

The V2 established huge gains in reliability and quality of construction over what went before. It was also the fastest Harley-Davidson to date. I have ridden them at high speed over long distances with nary a complaint. The Evolution has a great reputation with the riding public. The consumer overlooked its many inadequacies because it was so much better than its predecessors were.

## The Maligned Hemi Head

I constantly deride the hemi head combustion chamber

and its mating domed piston because of their design inefficiency. The old-style hemis like the Shovel, Pan, and Knuckle had high, round, domed pistons and equally deep, round-pocketed combustion chambers.

There is no incentive for mixing air and gas in combinations that can burn. The dome interferes with the sparked flame travel over and around the intrusive piston-top design.

Let us not forget the 1957-1985 Ironhead XL Sportster hemi head, which saw slight improvement in design as the sport hotrod Harley. The combustion chamber shape was the same as its Big Twin counterparts. However, the compression ratio increased, assisting in burn and power.

More importantly, there was no rounding of the piston dome, although it still extended far into its mating chamber.

The mini-flattop, one-dimensional pyramidal dome faced the valves. It was inefficient by modern standards, but was far better than the Big Twin variety.

The modern performance hemi head may look the same as the dinosaur technology hemi but has subtle differences that make all the difference, such as a modest dome and equally shallow combustion chamber. Furthermore, there is a circumferential angled squish band inside the combustion chamber extending a short distance up from the head gasket surface. Its mating squish area is the angled portion of the shallower dome as it protrudes up from the piston crown.

Larger-bore engines typically use this design. Hemi use is necessary where big valves will shroud the air-fuel mixture as it enters the convoluted, bathtub-shaped combustion chamber like that on the Twin Cam 88 and TC96. Bigger valves,

particularly the intakes, need room around them for the unimpeded flow not found in smaller, traditional squished chambers.

There is a squish band in modern hemi-heads incorporated into the beginning of the piston dome as it graduates away from the piston crown. It is not very noticeable because it angles on a similar plane to the doming and runs the circumference of the dome. There is a similar mating circumferential mating area at the combustion chamber as it extends into the head away from the surface area of the head gasket.

The piston dome is not as intrusive as the hemi piston of old as the dome is lower and not as pointed. The combustion chamber is not as deep as well. This is a huge improvement over the 1936-84 hemi heads but still not as good as, say, the higher-flowing Screamin' Eagle HTCC heads and piston top crown design.

## The Twin Cam Corrected Evo Defects

The Twin Cam addressed the plethora of Evo issues, such as porous engine casings with superb metallurgy, new manufacturing technologies, and vastly improved quality control.

Addressing other problems, such as chronic Evo weeping base gaskets, with simple but effective redesigns, became the order of the day. A simple hollow dowel now interrupts and channels the return-oil passage from the top end past the base gasket surface to continue into the bottom-end timing chest. The dowel protrudes up into the mating cylinder and down into the cylinder deck of the engine case. Moreover, an O-ring encircles the dowel to ensure no escape of errant oil is possible.

The mating surfaces of the cylinder base and the engine case decking are flat, straight, and smooth. Sometimes the cylinder deck face of both Evo engine case halves did not meet precisely on the same plane. This, of course, offers great opportunity for wayward oil to escape gasketing not designed to seal such an uneven surface.

H-D eliminated the base gasket altogether in favor of a superior methodology of using an O-ring juxtaposed between the base of the cylinder liner and engine case. Leaking base gaskets, so common with the Evo, have become outdated with the Twin Cam. It is extremely rare for a Twin Cam base to leak oil or even weep an oil film.

The H-D factories have steadily modernized over the years with state-of-the art procedures, techniques, and equipment. Now, robots tirelessly work side by side with a loyal and

enthusiastic workforce.

## Bill Davidson Introduces the Secret P22 Project

The covert project code named P22 was a well-kept secret indeed. One insider confided that he was nervous about talking in his sleep because his wife might overhear details of P22.

Buzz Kanter, Chris Maida, and myself, the publisher, editor, and Techline columnist respectively of American Iron along with some Cycle World staff, were the first outsiders to view the P22.

The P22 was code for the ultra-secret Twin Cam engine.

In 1998, when the Harley-Davidson Twin Cam design engineers were teaching the American Iron Magazine crew, including myself, about the new engine, Bill Davidson himself gave a heartfelt and emotional speech. We watched Harley history unfolding as Bill introduced the new Twin Cam engine just as his great-grandfather had done with the ‘36 Knuckle. Bill proudly told us, “When we think of Harley-Davidson, we think about its strong history, a history we are sitting in the midst of here in the [Harley-Davidson] Archives. I think about my great-grandfather and the guys who were on the end of the production line as they watched the 1936 EL come off the assembly line. For me to be sitting here, talking about Harley-Davidson’s future, it’s a great, proud moment for me.” 1

Bill Davidson then said that it was of paramount importance that Harley-Davidson stayed within tradition and maintained the H-D look, namely a V-twin configuration with pushrods and remain air-cooled.

The radical design change evolved around and from Environmental Protection Agency (EPA) dictates that forbade noise emitting from the timing chest spur-cut gears.

The Factory chose supposedly quieter timing chains to drive the cam lobes. Please conceptualize one camshaft with its four cam lobes like those in the Evo and predecessor Big Twins. The necessary length of a cam chain connecting the drive sprocket on the end of a driving pinion shaft to another on the end of the camshaft above it would dramatically change engine design and shape. The classic V-twin would unacceptably elongate.

Furthermore, the frame would need to change to house this taller engine. A frame change would lead to many other changes like gas tank shape, and on it goes.

This would be intolerable to the riding public, the Motor Company, and certainly to Bill Davidson.

Two cams, side by side with two cam lobes each became necessary to maintain classic looks. The rear cam was offset approximately to the rear of the timing chest.

The connecting drive sprockets and timing chain on the outboard side of the cam support plate powered the rear camshaft. The necessary length of the timing chain did not affect engine height because of its angular offset. The rear camshaft sprocket connects to the front camshaft sprocket straight across on an even plane via a second, shorter timing chain on the inboard side of the cam support plate.

To me, this is the strength and weakness of Harley-Davidson design all wrapped up in one. I, like most of you,

want this. I take it as strength. This is what I want to ride.

However, the weaknesses are glaring.

## The Weakness of Nostalgia

How can an engine maintaining its tradition survive the ever-increasing pollution dictates of the EPA without using new technologies to reduce both noise and exhaust emissions?

### The Tail Wags the Dog

I can trace every major engineering change to a Harley-Davidson since the circa-1980 introduction of the rear belt drive because chains were too loud to meet EPA emissions guidelines.

In fact, now that I am dwelling on this, many changes before the belt drive were a direct result of lessening emissions, whether they be pollutants or noise, such as the 1977-1978 debacle involving shortened valve guides, valve seals, and venting hoses running from the cylinder bases to the tappet guide blocks. I will cover this at great length in Volume VII: Shovelheads: 1966 to

Emissions must dictate the engineering, which, like it or not, should determine the look of an engine, within bounds, of course. This is easy in a car: the engine is covered and not part of the marketing aesthetics.

However, the tail is wagging the dog when nostalgia dictates the engineering.

Environmental Protection Agency Specifications for 2010

The watershed year will be 2010 for many vehicle manufacturers with Harley-Davidson's nostalgic design being particularly vulnerable.

Motorcycles above 280cc, referred to as Class III in EPA literature from 2006 to 2009 have difficulty, but it is not impossible, meeting exhaust emissions standards of no more than 12.0 grams per vehicle kilometer (g/km) of carbon monoxide (CO). One kilometer is a metric measurement that equals .621 miles or about ⅝ of a mile.

Year 2010 Emissions standards, thankfully for the survival of Harley-Davidson as we know it, do not reduce this amount. Class III 2006-2009 motorcycles, which again include all Harley-Davidson's, have a limit of 1.4 grams per vehicle kilometer of hydrocarbons (HC) plus oxides of nitrogen (NOX). Oxides of nitrogen define as the sum of the nitric

oxide and nitrogen dioxide contained in a gas sample as if the nitric oxide were in the form of nitrogen dioxide.

Gobbledygook for sure, but the bottom line is that this is bad stuff and is polluting our environment.

However, the kicker is that the 2006 to 2009 limit of 1.4 g/km of hydrocarbons plus oxides of nitrogen will reduce to 0.8 g/km in 2010.

How Can Present Twin Cam Design Survive?

The term later really means until the emissions dictates become even more stringent. Once technology helps meet one goal for mass production, the next ever-more-restrictive goal is set.

Therefore, I can now ask the rhetorical question. How can present design no matter how progressive survive future Environmental Protection Agency constraints? Design based on nostalgia is on the endangered species list!

I will not broach the subject of alternative power replacing oil, leading to a very new type of engine or a hybrid one, because the world is not economically able to extract itself from this interwoven oil product anytime soon. However, I suppose a hydrogen fuel cell's flat cellular structure could be designed as a V-twin-shaped shell to satisfy our need for the past. Surely, the fuel cell's one emission, drinkable water, is an attractive alternative.

The immediate reality for Harley is to play catch up with existing technology that has been around and used for decades by other manufacturers such as the Japanese. Of

course, Harley has tried new technologies with the V-Rod and, most recently, the more radical departure from tradition, the 2008 1125R Buell.

## AMF: Progressively Ahead of its Time

Actually, it was the much-maligned American Machine and Foundry (AMF), the bowling ball manufacturer that blessedly had a major hand in the survival of Harley-Davidson. In the 1970's, AMF made the most interesting and progressive radical departure to meet EPA dictates for 2010.

The choice faced by AMF, was whether to modernize the V-twin Shovelhead, whose legendary warranty problems were sinking the Factory relative to its competitors, or pursue a design ahead of its time, the code-named Nova. The Evolution, with a pushrod-operated V-twin air-cooled

motorcycle in a two-cylinder 1340cc design with a four-speed transmission, evolved through AMF's efforts, saving the day.

However, the Nova was an overhead cam (OHC), water-cooled V-twin, in various two- to six-cylinder designs ranging from 200cc to 1500cc. The Nova employed a five-speed transmission years before the introduction of the 1980 five-speed FLT Shovelhead. This was visionary but radically dangerous, considering the market during that era. The Nova would save Harley-Davidson with the EPA but would doom Harley-Davidson with the riding public back in the 1970's.

In a bid to placate the marketplace, the Nova's radiator hid under the seat, incorporating two large air scoops to redirect the cooling air to its heat-dissipating finning. I will explore AMF's twin strategies of designing the Nova through Porsche and developing the Evolution in great detail in Volume III: The Evolution: 1984-1999.

However, this design would easily meet basic 2010 dictates and probably the next round after that. Here we have the maligned AMF reaching into the future thirty years ago and using technologies that the EPA guidelines may force onto Harley-Davidson after 2010.

Will the Twin Cam survive the 2010 dictates?

I think so, but not the round after that.

Why do you think the Factory pushes the V-Rod so rigorously with lower prices and so many happening and progressive models replete with a separate drag-racing class? The V-Rod and the 1125R Buell will be the future of Harley-Davidson. They are radical departures for the not-so-distant future replacement after the Twin Cam's demise!

Maintaining customer-driven tradition becomes H-D's greatest challenge. For now, the Twin Cam is the king of the old technology and all that has gone before with the Big Twin models.

## The Strength of the Twin Cam

Rigidity is a pervasive and very important theme in the Twin Cam. The ability to have separate components act as one and in harmony allows for advances in handling characteristics and thus safety, efficiencies in power transfer to the rear wheel, vibration reduction, and increasing parts longevity.

### External Rigidity

Older Harley's flex and strain as power surges through the

powertrain.

External changes incorporate the ultimate torque arm. Speaking of torque arms, I believe that Carlini Design (www.carlinidesign.com) was the first to use them, with the Evo Dyna models. A torque arm was very necessary because the rubber mounting of the separate engine and transmission caused much destructive flex, particularly with performanced engines. The aftermarket torque arm attaches the engine to the transmission on the opposite side of the primary system. It connects the rear of the timing cover and the front of the end cover of the transmission. The torque arm stopped the Evolution Dyna engines from extreme rocking by providing extra rigidity to help the primary system in its thankless side job. All too often, the Dyna engine would rock so much that the bottom of the primary casing smashed into the lower frame member underneath. Physical parts damage, premature parts wear, inefficiencies in transferring power back through

the powertrain and bad vibration are all victims to the lack of rigidity! The torque arm stopped parts damage, extended parts life, added efficiencies to the powertrain, and was instrumental in reducing vibration caused by flex.

Harley-Davidson relied on the aluminum primary system attaching engine and motor to provide necessary rigidity to keep the two major components in line. The primary's job is to keep dirt away from internal components and to bathe them in cooling and lubricating oil, not to stop engine-transmission flex.

The Touring and Dyna rubber-mounted engine and transmission move together as one. They firmly attach so the engine and transmission approximate uni-construction with all the inherent benefits. The utilization of a uni-construction engine and transmission cases over the years for Factory race bikes and sanctioned racing events is no mistake.

In the 1950s, the uni-construction Sportster became king of the hill, dominating the hillclimb circuits, flat track, and oval racing. In the last few years, Factory-sanctioned drag racing features the uni-construction V-Rod in separate and special classes. The newly introduced uni-construction Buell 1125R in 2008 has huge potential for racing.

The Twin Cam Softail models also approximate engine-transmission uni-construction with a further refinement not possible on rubber-isolated powertrains. The main center frame downtube attaches to the top of the transmission casing

while the bottom of the transmission supported by the engine casing bolts securely to the lower- frame tube members. The engine and transmission are now part of the frame on the Softail models. Conversely and cooperatively, the frame is now part of the rigid unit construction of engine and transmission.

No more will parts counterproductively flex with strain as they have in the past.

Harley-Davidson is not resting on its laurels with its accomplishments, as the quest for improved rigidity is an ongoing work in progress.

Internal Rigidity

Rigidity pervades the internal Twin Cam design also, translating into superior handling and control, vibration

reduction, reliability, durability, longevity, and, finally, into one tough machine.

Internal rigidity enhances with shorter, beefier piston cylinders. As I have mentioned, the cylinder sleeves that house the pistons are thick at .247", which is about .075" thicker than its Evo counterpart is. The Factory reasoning behind this ¼"-thick sleeve liner is that it provides stability for the piston as it changes thrust axis, particularly at the bottom of the stroke. As the piston descends in the cylinder, it must reverse direction at the bottom and ascend the bore again. Since the cylinders are at a 45-degree angle to each other, the piston pushes against one side of the barrel going down, but when it changes direction, it thrusts against the opposite side. The ongoing theme of rigidity in this platform engine comes into play once again, providing stability for component parts. In this case, inflexibility is due to the thickness of the liner and the new spiny lock iron sleeve cast

into the high-pressure, die-cast cylinder, replacing the predecessor separate pressed-in cylinder sleeves. This thicker cylinder liner and construction remain the same for both the TC88 (4.000” stroke) and TC96 (4.375” stroke) engines. The TC88, TC95, TC96, and the TC103 engines use the same cylinder. The liner thickness allows the TC88 and TC96 engines to be bored from their stock bore of 3 ¾” to 3 ⅞” to achieve 95 cubic inches and 103 cubes respectively.

Two major crank changes in recent years are the Twin Cam crankpin and attachment style and the late-Evo innovation of one-piece flywheel halves and integral shafts cast and machined as one. Flywheel integral shafts: the sprocket shaft and its left-side flywheel along with the right-side flywheel and its pinion shaft promote internal rigidity and precise vibration-robbing fitment. Robotically machined trueness ensures no runout is possible between shaft and flywheel in the one-piece shaft-flywheel component.

Previous model separate shafts truing as precisely as possible to perpendicular to its mating flywheel half were held securely with Loctite on big pinion nuts to massive sprocket shaft nuts. In most cases, a screw-held half-lock ring further ensures no nut loosening. The reader can easily discern which methodology is the more rigid and less likely to induce vibration. Separate tapered fitment shafts and flywheel halves are fraught with potential misalignment and flex issues.

The Twin Cam's unique straight crankpin has no tapers like previous Big Twins and all Sportsters. These have crankpin tapers that can allow flywheel halves to shift either in the truing process or under extreme duress. Do not get me wrong, as tapered separate engine shafts and crankpin are a proven methodology in the hands of a skilled and experienced mechanic. They have served the industry and rider well for many decades.

I have taken the trouble – actually, it was fascinating -- to disassemble a Twin Cam crankshaft. On earlier engines, it is easy. Zap off the crankpin nut on the sprocket shaft side and gently tap the flywheel half on the side below as far away as possible from the crankpin with a mallet made from hard rubber, soft metal lead, or brass. The flywheel separates easily from the tapered crankpin. Well, it is not so easy with a Twin Cam.

The biggest diameter crankpin and never-seen-before straight crankpin press-fits into each flywheel half at .007" interference fit. The flywheel halves are not easy to separate. In fact, it is quite a job compared with earlier cranks to disassemble. A hydraulic press is necessary to separate the flywheels, which is a long way from a tap with a mallet.

The Factory in some service bulletins, notably TT324, Twin Cam Crankshaft Inspection and Run-Out Specifications (July 26, 2007) and TT352, Twin Cam

Crankshaft Assembly Run-Out Inspection (December 3, 2007) along with two short videos are teaching technicians about flywheel alignment boondoggles. Some of these young guys have never even trued a set of flywheels or taken apart a Twin Cam crank. The bulletins say that traditional tapered crankpin and shafts out-of-true conditions can exist with Twin Cam I do not get this. Apparently, the flywheel halves can cock inwards opposite the crankpin. In addition, it seems that they can spread outwards, again opposite the crankpin. Thirdly, the flywheels can skew sideways so the crankpin is no longer parallel with the engine shafts. Lastly, the flywheels can scissor if one flywheel moves ahead of the other. Understanding scissoring is a no-brainer. One flywheel will move forward or backward on the crankpin.

Comprehending the first three conditions is easy on predecessor flywheel assemblies where shafts and crankpin have male-tapered ends that fit into female tapered holes. The flywheels can easily shift inwards, outwards, or skew in tandem sideways on their tapers.

However, how does a straight, hardened crankpin allow inward and outward shifts or tandem skewing when either male end interference fits into smaller straight holes in the flywheels? If this does indeed occur, it seems to me that it could only happen in the initial assembly process at the Factory.

Pressing the crankpin under great pressure off-kilter into its smaller mating hole would be necessary. Material would unavoidably shave off on the side of the hole to allow a misaligned fitment. If this were the case, compromising the

.007” press interference could easily allow scissoring of the flywheels to follow. One would think that the straight fit of the crankpin into each flywheel half would not allow this. I will examine this in detail in Chapter V: subsection, Twin Cam Flywheel Misalignment.

The Factory is now indeed encountering a problem beginning in 2007 and carrying over into 2008 with flywheel halves scissoring out of trueness alignment on the TC96 engines, and the CVO 110-cubic-inch mills. This blows me away because I know how rigid and together TC88 cranks are.

There are two possible scenarios here other than the Factory’s initial incorrect assembly already discussed. One, the increased torque of the TC96 and especially the CVO 110 over the TC88 is causing this malady. In fact, this is the case whether alone or in concert with another factor. I cannot help

thinking that the Factory, for whatever reason, maybe an easier and therefore more cost-effective truing process, has reduced the .007” press interference fit to a lower value. There will be much more on this again in Chapter V: Vibration.

Harley-Davidson warns repeatedly that Twin Cam flywheel assemblies are not rebuildable. The Factory enforces this dictate through warranty rules and not making available any rebuild component parts.

However, there are those of us who like a challenge and love to rebuild the un-rebuildable. As a teaser, many in the automotive industry repair scissored flywheels first by re-truing them to close tolerance specifications and then by tack-welding the crankpin end to the flywheel. Many times, this is a standard scissoring preventative procedure with cranks used in high-torque engines for racing.

I have digressed and perhaps lessened the reader's insight concerning the rigidity of the Twin Cam flywheels that assist mightily the smooth operations inside the engine.

I will strengthen readers' insight by concluding that the crankpin is huge compared with the past. The connecting rods are the strongest ever. Their muscularity rivals and sometimes exceeds the big-inch engine rods available from the aftermarket. The increased number of captured bottom-end bearings is longer than their predecessor equivalents. The piston wristpins with their corresponding small-end con rod bushings are the biggest and strongest ever, and, finally, the shafts are integral with their flywheel halves.

For the most part, the Twin Cam flywheel assembly is the best built, truest, strongest, most durable and rigid assembly

in the history of Harley-Davidson.

## Machining Quality

The machining is far superior to what went before. For example, the machining of the Twin Cam engine case base gasket deck surfaces is with the two case halves bolted together to provide a uniform fit. I have heard reports that the machining of some year Evo casing halves occurred separately in different locations, so that sometimes the mating gasketing areas did not match planes with each other with resultant leakage. I have already said that the Twin Cam casings are only clamped five times, reducing machining distortions from the previous thirty-two clampings necessary for producing Evolution casings.

As testament to their quality, only two companies, S&S Cycle and JIMS, in the aftermarket make replacement casings

for the Twin Cam.

The public may purchase S&S Alpha casings. Their purchase is available through any S&S Cycle dealer like my shop, Heavy Duty Cycles in Toronto. A dealer locator is on the S&S web site.

Both offer excellent quality.

However, the purchase of JIMS Twin Cam casings and big-inch Twin Cam kits must be through an authorized Harley-Davidson dealer. It gets better as only licensed All Harley-Davidson Drag Racing (AHDRA) racers are able to purchase just the casings.

Casting Improvements

Again, the excellences of the Twin Cam engine casings are fundamental works of art when compared to some-year porous, poorly machined engine seal holes, and weak and cracking Evo casings.

As of this writing, Twin Cam casings have been in use for nine years with nary a case of breakage, leaking, porosity, or any other type of concern. The high-pressure, die-cast aluminum halves split down the middle as always, promoting rigidity, strength, and durability as the framework for what is inside and on top. Designed especially for these cases, the specially formulated proprietary 360 alloys imbue visual and tactile quality that is evident as soon as you look at or touch them. No more malfunctions, leaks, or breakage.

There is more. As further testimony to the quality, strength, and durability of Twin Cam engine casings, we at

Heavy Duty Cycles routinely and safely build 116.2 cubic inch Twin Cams using bored-out stock engine casings to 4.060” and 123.6 inchers taking the case spigot size out to 4 ⅛” (4.125”) with the same 4 ⅝” stroke. In practice, we round these engines off to 116” and 124” respectively.

Ninety-six cubic inches with a 3 ⅝” bore is the street-safe limit on most year Evos although the weak engine castings prevalent from 1989 to circa 1993 even made this increase a dangerous proposition.

The Harley-Davidson 110-cubic-inch engine uses a bore of 4.000”. An extra .060” increase in bore would have allowed its power plant to become a 116” ground-pounder. I will look at these engines thoroughly in Chapter XII: Cubic: Strokers and subsection, 124 Safe Cubic

For the most part, Twin Cam cast flywheels with integral shafts are on a level with the best in the aftermarket. Twin Cam cast heads and barrels are superlative in design and quality. They will last many hundreds of thousands of miles and then some. The cast plus 12% silicone heat protected hypereutectic pistons are the best ever with the closest fitment. They are going to set mileage records never seen before with H-D pistons.

Former 12% silicon content eutectic Evo pistons were good. However, they cannot hope to match the durability and longevity offered by the Twin Cam pistons. Old style--less than 12% silicon--hypoeutectic pistons are not even in the current ball game of superiority.

The cast cam-support-plate is very efficient as an oil-routing system and ably handles the outboard cam-journal support function, relieving the timing cover of this onerous job.

There are other ongoing examples of casting improvements but enough said.

## Metallurgy

Metallurgy in the casting, forging, and machining processes is a given. How could components be so good without the best-quality metallurgical materials? I must say the Motor Company excels in using the best materials available.

## Parts Quality

Parts quality shares much room with casting quality, as sometimes they are the same. I will list a few examples not

already reviewed elsewhere.

Automotive hydraulic lifters are a good example of longevity, efficiency, and quality.

First used in the Evo, they had great success, especially when compared with the fragile Shovelhead and Panhead hydraulics. The lifters are refined to produce longer life. The lifter tappet rollers--again, first upgraded in the Evo years--are now even better.

Helical transmission gears will prove quieter than traditional spur straight-type gears as they are always engaged, negating any backlash noise.

Even the powdercoating process is tough as nails. Engine components are powdercoated wrinkle black or blond as the

case may be after casting or molding but before machining. Think about it. This powdercoating is tough enough to withstand the various clamping and machining operations. If marked with one small cosmetic defect, the part will not pass quality control, resulting in disposal. This powdercoating is very much superior to what is available to us in normal powdercoating shops.

Another great example is the gasket quality and improvement with the Twin Cam over the Evolution, whose gasketry and sealing were superior to the Shovelhead but nothing compared to what followed in 1999. The Twin Cam gasket and sealing surfaces they join assist immeasurably to the integrity and quality of the much-improved system.

Gone are defective or weeping gaskets.

In rare cases, such as the CVO 110 engines leaking oil from

the rear head gasket, it is not the gasket's fault but excess heat above the already onerous emissions Stoichiometric air-fuel ratio of 14.7:1. Rear cylinder temperatures are overbearing as air-fuel ratios reach from 14.8:1 to an unbelievable 16.0:1. The cylinder liner seeks to separate from its cylinder. Expansion forces come into play and the best gasketing ever is overwhelmed.

Let us take the rocker gaskets, for example; they are rubber-coated with an embossed steel core providing uni-pressure throughout. The embossing is a ridge, if you will; something like the silicone beads now found on high-quality aftermarket gaskets except it is a coated metal. This metal embossing follows around the boltholes as well as the gasket surface to provide a uniform pressure seal on quality machined surfaces.

The reader must understand that I, like you, have been suffering from Harley gasketing oil leaks for thirty years now. The cure is here. No longer does Harley mark its spot. Well, not from leaking gaskets anyway.

or oil are sometime synonymous words for oil expelling from the engine through the breather venting process that is supposed to separate air from oil spills out the breather onto the side of the engine. H-D can still mark its spot via blowby aggravated by bigger swept volumes as in more cubic engines and/or sustained high-rpm riding.

However, gaskets and O-rings are almost foolproof now for the first time in Harley-Davidson history.

Engineering Design Quality

There are many examples, but I think I will lead off with a

simple but brilliant design change.

All previous engine covers on Harley-Davidson Big Twins--and current Sportsters, for that matter--use engine covers that do double duty as load-bearing components. All engine covers like the nosecone or timing chest cover, the split two-piece rocker wafers and the guide block covers are just that, covers. Covers acting as load-bearing supports for other engine parts became a thing of the past beginning in 1999. Evo and former engine covers fulfilled the dual-purpose roles of attempting to seal oil leaks while acting as outboard moving parts supports. In the Evo, the nosecone or timing cover supports the rotating pinion shaft and cam outboard journal.

The V2 lower rocker wafer supports the rocker arm shafts as the rocker arms transfer the dictates of the rotating elliptical cam lobes, opening the valves and then allowing

them to close. The compressed valve springs force the valve back into its seat, thereby pushing the valvetrain back down following the descending cam lobe.

The Evo guide block holds the tappet assemblies as they push the valvetrain up and are pushed back down by uncompressing valve springs. In the Twin Cam, a cover's function is to seal the oil inside the engine and secondarily do double duty as a noise suppressant.

Covers that support moving parts will often transfer noise emissions outside the engine. It is still noisy inside the Twin Cam engine, but more of it is contained inside. Inside the Twin Cams' internally noise dampening webbed and finned rocker wafers is a separate support plate for the rocker arm support. This plate bolts directly to the head through the lower wafer. It also holds the breather apparatus, which consists of the oil separator, the flapper valve, and the scrub sponge, which I will be discussing in Chapter XII: Cubic

Inches: Strokers and Big subsection, The Many Cures for Puking

Inside the larger Twin Cam nosecone, cam, timing, or timing chest cover--take your pick of many synonyms--is a cam support plate that also directs the oil through the engine from the internal geroter as opposed to the previous gear-drive oil pump. The cam support plate also supports the two cams on the outboard journal side as well as the end of the pinion shaft that passes through the plate to hold and drive the rear-cam pinion-drive-gear.

The pinion shaft also accepts oil from the oil pump via the distribution support plate into the shaft for internal delivery to the heart of the engine, comprised of the crankpin, con rod roller bearings, and the connecting rod big-end races. The busy and multifunctional cam support plate does not interact

in any way with the engine timing chest cover. The plate firmly bolts to internal and integral cast appendages of the right-side engine case.

The engine case supports and directs the tappets instead of using removable guide or tappet blocks. Chrome covers fit over the engine case tappet support holes. I have not been a big fan of eliminating support components like tappet blocks since they are rebuildable and replaceable. These relatively inexpensive support components protect and increase longevity of very expensive apparatuses, such as engine case sets. Now we face the prospect that the engine block tappet holes, once worn out, require engine case replacement. However, it looks to me as if there is enough room to accommodate an oversize tappet. The quality of the aluminum is very much superior to that used in the Evo, which will stave off problems, as will the cleaner filtered oil that this engine utilizes. Finally, nine years after this engineering introduction, I see no evidence of wearing tappet holes.

Old-time mechanics like my mentor Fast Eddie, who we respectfully called Mr. Harley-Davidson, warned me that the next Twin Cam engineering innovation, the internal oil pump, had already failed in the early years. He said that engine debris in the oil ultimately destroyed these early internal oil pumps.

Speaking of Fast Eddie, it is now an opportune time for another Donny's Blasts from the Past digression.

Donny's Blasts from the Fast Eddie and the Land Speed Record

My old pal Fast Eddie, the super-wrench of his day, set a land speed record at Wasaga Beach north of Toronto on a Snortin' Norton shortly after the conclusion of World War II.

Mouldy ran the local dealership where Fast Eddie wrenched. The oil-soaked wooden floors smelled wonderful. His workers called him Chief to his face and Mouldy when he was not around to hear. Mouldy was badgering Fast Eddie to ride a souped-up Knucklehead on Wasaga Beach, north of Toronto.

Wasaga was the definition of cool. It was a biker hangout from the ‘40s to around 1990 until the good people with money took over. They made the beach a respectable but very boring place with far too many rules.

Mouldy wanted Fast Eddie to make an attempt at a land speed record. It was too windy for high-speed riding on loose sand. Eddie refused; he was good and he knew the Knuck’s limitations. Mouldy coerced a hapless soul to make the attempt. A high-speed wipeout claimed the ill-fated rider’s leg.

It was not that Mouldy was a villain, because he was a true enthusiast. The reader should understand that this was another era. Post-World War II was a macho world where men were men and women were double breasted. Of course, the word macho was unknown.

Oh, the rider who lost his leg; well, Pegleg was back at Wasaga on his rigid frame the next summer. The guys were fun but tough and didn't give an inch.

I had three friends back then named Tim. We called one Timmy; the other was nicknamed Two Foot and the third, One People were always trying to figure out if the third Timmy had two legs or one.

Anyway, Eddie was pissed. Bikers had wonderful ways of

getting even. Fast Eddie, jumped on a Snortin' Norton and set a land speed record on an English bike called Limeys back then for Mouldy's H-D dealership! By the way, Mouldy's main competitor down the street was the Norton dealership.

Anyway, a retired Fast Eddie, who had slowed down considerably, dropped by my shop, Heavy Duty Cycles, shortly after the introduction of the Twin Cam. He came by every six months or so, to give me an H-D tool from the past, like a pre-1955 pinion race-lapping arbor that he could not bear to give up before.

"Is it correct that Milwaukee has moved the oil pump inside the engine?" this true enthusiast queried with a worried look. "Yes, Eddie," I replied. I went on to explain the details as the Factory engineers had taught me. "Never mind that, Donny; figure out a way to fix them when they break down.

Harley tried an internal oil pump before World War II. The engines all failed."

While the Factory has had minor issues with the Twin Cam oil pumps, they have been an engineering success this time around, as long as other parts do not screw up.

Another simple but wonderful engineering innovation, filtering oil before it goes into the engine, not afterwards as on all previous Big Twins. The Shovel camel oil filter removes small rocks from the dirty oil while the Evolution adapted a 30-micron (30-millions-of-a-meter) pleated paper filter. Early Twin Cams from 1999 to circa 2006 filtered oil with a 10-micron synthetic media-pleated filter. H-D gets very serious about keeping the oil clean with a 5-micron synthetic media oil filter for all 2007 to present models and 2006 Dynas (black #63731-99, chrome #63798-99).

Strangely, both 10- and 5-micron filters have the same part number so it is difficult to tell exactly when the increase in filtering was introduced.

A 5-micron filter will collect a percentage of particles bigger than 5-microns from the oil before it enters the engine. My definition is deliberately vague because different authorities and filter companies employ varying standards to define filtering capabilities.

However, the reader needs to know one thing and that is filtering on Harley-Davidson's have made huge advances.

Early and Late Twin Cam Oil Pumps

Pictured on the left is the 2006 Dyna and 2007 to present Twin Cam oil pump, while the one on the right is the basic pump used from 1999 to 2006 (except the 2006 Dyna). All previous oil pumps in use by Harley-Davidson, except the 1977 to present Sportsters, used gears to pump moderate

volume and moderate pressure. The geroter-style oil pump impels higher volume and higher pressure than gear-driven ones. Oil feeds into spaces between the geroter nubs. Oil entrapment occurs as the rotors turn. The rotating nubs force the oil into an increasingly smaller space before opening a passage for it to escape, creating the pressure and volume. The 2007-style pump also creates more oil volume and pressure than the earlier Twin Cam pump because the geroters are thicker and there are fewer nubs. Fewer nubs allow more oil to enter thus entrapping a greater amount of lubricant. The 1999-2006 pumps have a twelve/thirteen-lobe interaction. The outer geroter has thirteen internal lobes while the inner geroter has twelve external lobes. The new 2007 pump has ten/eleven nub rotors so 10% more oil feeds into the engine. More oil (23%) removes from the timing chest and returns to the oil tank. H-D hopes this will control blowby, as not as much oil will be available to carry up into the rocker boxes, theoretically alleviating the danger of the breather apparatus being overwhelmed. In my opinion, this definitely helps. An

easy way to identify the pumps is on the return side housing (shown). The right side early pump has a round oil-return hole while the new pump has an elongated hole on the left side as viewed in the picture. The holes align to their respective cam support plates

Filtering and other changes have allowed the Factory to extend service life between oil changes up from five hundred miles initially to one thousand miles and then every five thousand miles thereafter.

A simple but effective engineering improvement, such as oil delivered to the bottom-end cavity, is there for the express purpose of lubrication for specific components. Oil from the top end now empties into the timing chest, not the bottom end like previous Big Twins. This prevents excess oil from sloshing around on the flywheels, causing inertial drag.

The new oil pump is a twin geroter, dual scavenge, crank-mounted and driven internal dry sump oil pump. Geroters are lobed gears that run inside each other.

A wet sump, never used in any modern Big Twin or Sportster, stores its oil inside the crankcase, while a dry sump stores oil in an external oil tank. A wet sump does not have a return side since there is no reservoir to ship the oil to, so a wet-sump engine bathes in the oil supply. A dry sump must have a return side to the oil pump to deliver the oil back to the tank.

The gear pump like that used on Evolutions, Shovelheads, Panheads, and Knuckles is okay but not the best. A gear-driven oil pump produces only moderate volume and moderate pressure. The geroter pump, which uses rotors instead of gears, produces high volume and high pressure so from this perspective is very much better than Evo and earlier pumps. Earlier oil pump gears mesh with each other side by side, but on the Twin Cam an inner rotor sits inside an outer rotor squeezing oil between the two to produce pressure and flow.

The term twin geroter is a little misleading since there are four rotors: two for feed and two for return. It actually means that there is both a feed and a return, which is necessary for a dry-sump system. The thicker geroters are for the return side while the skinnier ones are for the feed and fit right over the pinion shaft.

- Crank-mounted signifies that the oil pump mounts on the pinion shaft. The rotating pinion shaft drives the inside rotors with the oil pump housing acting as a shell. It bolts to the inboard side of the stationary cam support plate.

- Dual scavenge denotes that oil is picked up for return to the oil tank from two places, the crankcase and the timing chest.

- Internal obviously means that the pump is inside the

engine, not outside like most predecessors Harley's.

A newly modified oil pump introduced with the 2006 Dyna and carried over into all the 2007 and later Twin Cams has a different porting arrangement than earlier ones. The new pumps will not correctly mate with 1999-2006 (except 2006 Dyna) cam support plates.

The new oil pump (26037-06) is superior to what went before. The original 1999 oil pump (26035-99) has gone through various improvements (26035-99B) through the years, but the pump has remained essentially the same. The 26037-06 oil-pump squeezes out 10% more oil on the feed geroter set.

Perhaps its biggest benefit will be in attempting to control oil carryover also known as or external Blowby occurs when

an oily mist, and sometimes straight-oil, will expel from the top-end breathers into the air cleaner and out onto the side of the engine. The 2006 Dyna and 2007 to present Big Twin oil pump has a 23% improved scavenging (return) capacity. Sadly, even with this extra scavenging, some bikes still suffer from oil carryover and mark their spot. Essentially, every attempt at controlling this malady since the 1980's, with major changes in 1992, 1999 and in the 2006 Dyna, carried over into all 2007 to present Twin Cams, while decreasing the incidence of the problem did not eliminate it. These attempted improvements have still failed to eradicate totally the oil's dripping and puking to the outside world.

The 1999-2006 geroter pairs are narrower. The feed geroters are about ¼" thick, while the ones on the return sides are double that. The 2006 Dyna and 2007 to present geroters are marginally thicker. One rotor sits within the other with a total circumference of about 1 ¾". The outside rotor is round

on the outside perimeter. The hollowed-out center has internal nubs much like a belt drive belt. The hollow, smaller inside rotor is partially round on the inside with two opposing flats that mate with and sit on the pinion shaft drive. The outside perimeter has external nubs that interact with those on the inside of the outer rotor. The mating nubs move in a seemingly eclectic but steady pattern because the inner geroter has one less nub or lobe than the outer. The meshing lobes open to let in oil and close in on each other, squeezing trapped oil out to make pressure.

Pumps from 1996 to 2006 (except 2006 Dyna) have a twelve/thirteen lobe interaction. The outer geroter has thirteen internal lobes while the inner geroter has twelve external lobes. Therefore, the new 2006 Dyna and all 2007 and later Big Twins, ten/eleven nub rotors feed more oil into the engine and also remove much more for return from the timing chest to the oil tank. The design seeks to remove more oil to prevent too much oil carrying into the top end by the

voluminous air displaced by the engine's descending pistons.

This is of particular concern to the high rpm and big-cubic-inch-crowd. This riding style and the increased piston swept volume of larger displacements severely exacerbate blowby.

However, the oil pump depends on a very true flywheel assembly with runout certainly less than .004" and preferably the old standard of .001" or less. Runout can hog out the pinion shaft bushing in the cam support plate that directs oil under pressure to the bottom end. Oil pressure depends on close fitment between pinion shaft and bushing.

Severe runout with scissoring flywheels primarily in the TC96 and CVO 110 engines has broken geroters, destroying the oil pump. I will be discussing in detail about the ramifications of the flywheels and shafts that have high

runout figures in Chapter V: subsections, Twin Cam Flywheel Misalignment, Left Crank Runout, and Right Crank Runout.

I have one major criticism of engineering in that the timing chain design for 1999-2006 Twin Cams turned into a debacle, which I will deal with later on in this chapter in subsection, The Achilles's Heel of the Twin Cam.

However, the great thing about the side-by-side dual cams is the engineers were able to eliminate an aggravating angularity of exhaust pushrods on Evo and predecessor engines. Twin Cam pushrod-angle holes are almost parallel.

Although there are more examples of engineering excellence brought into the Twin Cam platform, I will end with another simple yet effective change to the oil reservoir.

The oil in the Evo pan under the transmission has a feed and return at the front which means that the same oil gets used over and over again, while fresh oil is allowed to languish at the rear. Oil breakdown through oxidation occurs with the overused working oil resulting from higher temperatures. Furthermore, pollutants do not disperse evenly throughout, preventing dilution. The Twin Cam internal design eliminates this situation by utilizing a baffle plate in the oil pan that ensures the returning oil follows a tortuous path throughout to ensure complete circulation. This minor change promotes lower temperatures and superior filtering through using all the oil all the time. The tank reservoir and even better baffling updates again in 2009.

Even smaller changes evidence the Twin Cam's determination to resolve all issues of the past. In 1999, the three sometimes leak-prone clutch derby hold-down primary screws on the Evo replaced with five screws on the Twin Cam. Furthermore, engineers replaced the previous good round

rubber O-ring of this primary clutch inspection derby with a superior quad seal.

These and other changes combine to form the platform for performancing modifications that have proven to be extremely reliable.

Manufacturing Quality

Without duplicating what I have already written, these chapters saturate with the quality of the Twin Cam and the majority of its component parts.

I will give one further example. Transmission gears and cams are heat-treated in the traditional oven-quench format as well as being heat treated by modern wave technology where in a mere 20 seconds a cam becomes red hot and then quenched to produce a .040 thickness of the desired hardness.

A relatively new technology to make gears quieter that did not even exist twenty years ago helps the Factory combat the ever-increasing noise reduction dictates of the EPA. Old-style, straight-cut spur gears clattered away as they interacted.

Now, machined full-contact ratio spur gears roll into each other, utilizing the whole contact area available. This disperses pressure and produces less noise. The problem arose after heat-treating the full-contact gears. The heat-treating causes some distortion to the machined surfaces, resulting in noise-generating high points. The new technology allows machining the gears again after the heat-treating process without damage. These are the quietest spur-type gears available.

People, Skill, Money, and Commitment

Many factors combine to catapult Harley-Davidson to the highest levels of manufacturing excellence.

Four areas must join for excellence: people, skill, money, and commitment. It all starts with a dedicated and skilled labor force. Management skills are evident everywhere within Harley-Davidson. I mean, just check out H-D's marketing capabilities. The workforce from assembly line workers to management has a love for and a dedication to their product.

Next, an adequate budget is always ready to supply the best tooling and facilities.

Equipment and machinery are constantly upgrading at H-D's extensive facilities so they will be ready for growth that is more productive. Robotic work centers tirelessly and accurately perform their specific jobs day in and day out. Vibration-sensitive tooling isolates from all interference by

operating on different platforms. Even the vibration from passing tow trucks isolates away from the machinery.

Organizational skill is even evident in the simplicity of distribution of supplies and components at the Capitol Road plant. Materials come in and the finished product leaves from loading docks on three sides of the manufacturing facility. Products enter and exit seamlessly without disrupting the ongoing manufacturing process.

## Quality Control

Quality control is an integral part of the manufacturing process from metallurgy, new technology materials like carbon fiber, engineering design, advanced robotic tooling, and a highly skilled workforce from the plant floor right up through management levels and top company leadership.

Computer-generated problem sourcing looks at warranty claims, parts failure, and complementary juxtaposition of components and how well they work together.

A quality control computer decides how many assembly line components need examination for further quality-control procedures and possible corrective redesign until potential problems eliminate.

Sophisticated light measurement instruments in sealed clean rooms measure parts machining in an instant and with remarkable accuracy.

Cold running testing of newly assembled Twin Cam engines on a bench where all types of probes, including vibration detection ones insert to measure and detect any problematic irritations.

I will conclude with another Donny's Blasts from the Past to demonstrate just how far the Motor Company has evolved.

Donny's Blasts from the Motorcycle Sabotage

One day back in the '70s, I was hanging out at Mouldy's dealership in the service department, as it was as close to heaven as I could get in my early days.

Mouldy was going berserk because, if memory serves, every one of sixteen FL police bikes blew the rear head gasket during predelivery inspection. To compound the problem, the gasket blew by the most difficult head bolt, which shrouds by the starter solenoid, heavy starter cables, and more delicate wiring.

The police wanted their bikes, and Mouldy only stocked up to three of each part.

When I tried to buy a part, he would say, “I have one for the police, a second for my good customers, and if I have a third, you can have it, Don.” I never did reach good-customer status, even though my full paycheck went to Mouldy every week for parts for my 1966 Shovelhead. Wife 1 and I lived and partied on her check because there was no party if my bike was broken down.

The mechanics could fix fifteen of the sixteen bikes right away; over three days because Mouldy had three separate head gaskets, three top-end gasket kits, and three gasket sets in stock. Each gasket set includes two head gaskets.

Fast Eddie was trying to placate the Chief -- which was Mouldy's nickname when speaking face to face with him -- when more news arrived from Fats. The gaskets had blown because the head bolt was missing on every bike.

Mouldy went nuts.

Fats began to pillage head bolts from two of the bikes on sale at the small dealership.

In those days, the owner of the dealership had much power. He was the only game in town, bikes always broke down, and they needed constant maintenance. There was no aftermarket. Back-alley shops existed, but they all depended on the sole dealer for parts.

Another time, Fats motioned me over to show me the inside of a wooden crate used back then to transport bikes

from Milwaukee to the dealerships. Silently, I read a scrawled note on the wooden plank. The scawling read, “Transmission Fucked.” The mechanics pulled the tranny and took off the handshift top. They found a ⅝”-wrench jammed inside the transmission!

That is just the way it was since the bikes were handbuilt, parts were laboriously handmade on lathes or milling machines, and somebody was always pissed at something.

## The Platform

The Twin Cam bore is 3 ¾” (3.750”) for both the TC88 and the TC96, which is a big increase over the Evo’s 3 ½” (3.500”) bore. The stroke decreases to 4.000” from the Evo’s 4 ¼” (4.250”) on the TC88 and increases to 4 ⅜” (4.375”) on the TC96.

The Twin Cam stock bore increase of .250” (¼”), over the Evolution does not add any stress to engine operation. In fact, it lessens. The reason is heat-producing piston speed decreases. Bigger bore is the safer way to more cubic inches.

However, the stroke in the Twin Cam, which defines how far up and down the cylinder that the piston travels, decreases a full .250” (¼”) from the Evolution in the TC88 and increases .125” (⅛”) over the Evo with the TC96 configuration.

As I have discussed, the improvements in casting qualities, parts design, parts excellence, engineering superiority, metallurgy, improved materials, and quality control all join to allow the safe use of mega-bores and longer strokes that were not possible in the past. This is big news for the performance set. The S&S 124-cubic-inch behemoth with a secure stroke of 4 ⅝” (4.625”) and a mega bore of 4 ⅛” (4.125”) builds easily and safely using the superb stock Twin Cam engine casings!

They are fast, and they hold up because of two overriding factors, the quality of the S&S parts and the quality, strength, and durability of the stock Twin Cam engine casings.

Now, the industry is building bigger engines using aftermarket engine casings with wider cylinder spigot-hole walls allowing the safe use of even bigger-bore cylinders. JIMS has a 131” Twin Cam using an even more reliable 4.500” (4 ½”) stroke and 4.3125” (4 bore that is only available for purchase through authorized H-D dealers.

I believe we will see much bigger-inch Twin Cams in the near future. The Twin Cam engine has been developed to run stock for much, much longer than any predecessor Big Twin.

A few serious glitches like the timing chest components require salvation from the aftermarket. A performance

obstruction with low head airflow manages easily with porting and polishing techniques or with head replacement for bigger engines.

This is why H-D engineers call this engine a The serious performance mechanic can use this structurally solid base for wild torque and horsepower increases in a variety of safe ways.

Therefore, if your heart desires, the Twin Cam staging area for more power is available, and it is fast, resilient, and reliable.

## The Achilles's Heel of the Twin Cam

Noise pollution and exhaust emission restrictions drove Twin Cam design.

Controlling exhaust emissions creates heat and therefore a cylinder detonation concern.

This ever-increasingly serious issue of emissions-control will culminate with new EPA guidelines in 2010.

However, it was trying to lessen noise that primarily drove Twin Cam design. The spur straight-cut timing gears used in previous models like the predecessor Evolutions and all that went before them were far too noisy to pass auditory emissions testing. Noise generates from three main sources: the exhaust, gears and, believe it or not, air rushing through the intake manifold. There are decibel limits that the bike must not exceed in drive-by testing. The combination of the three main generators of noise cannot exceed this limit, which in 1999 was 80 decibels.

The market driven need for a throatier exhaust tone can be louder if gear and manifold noise reduces. The reader will see where I am going with this since the majority of Harley riders want at least a little more exhaust noise. As always, consumer wants drives for a particular auditory exhaust chugging potato-potato that is music to our ears.

However, the gear backlash noise was the sleeper. Mechanical noise reduction in the engine meant changing from the traditional spur gear meshing to sprockets and a silent plate chain. Metal expansion from heat and contraction became an insurmountable problem without dramatic change. The Evo and earlier model gears whined when tight and clattered when loose.

The design sequence of events necessitated the move away from the gears to the silent chain method to achieve noise reduction. This produces a consistency in sound as well as the desired reduction in noise.

A chain with two sprockets takes up much more room than two meshing gears. In this case, we have engineering driven by aesthetics. If a single chain went straight up from the pinion drive gear to a one-cam location, it would raise the engine casing's deck too high. Engine shape would change too much. We all know that this would not be acceptable from a marketing point of view. Therefore, two chains with two cams are necessary because the first or primary drive chain would need to offset to maintain traditional engine shape.

The primary silent drive chain offsets from perpendicular about 22 ½ degrees to allow for the desired deck height of the engine cases. This primary chain connects the driving pinion shaft gear to the rear-cam-drive-sprocket. A second chain and cam allows centering of the cam lobes under their respective valvetrains, which gives mechanical efficiency.

Aesthetically, the pushrod tunnels remain in essentially the same places as before, mercifully with the exhaust pushrods not severely angled as on predecessor motors.

Therefore, the two cams with two lobes each evolved instead of one with four cam lobes with the added mechanical benefit derived from straighter angle pushrods.

All of this for a traditional and nostalgic looks!

The TC88 dual cams connect via integral sprockets and by a secondary silent chain behind the cam support plate with the rear cam driving the front cam. The cams are beside each other and sideways parallel.

Both TC88 chains have spring-loaded tensioners for constant and supposedly consistent adjustment. Factory engineers told me they expected these adjusters to outlast the engine. However, the problems began right away in 1999, and I document them extensively in Volume I: The Twin Cam in Chapter II: Teething Problems.

## Early Timing Chest Problems

The early 1999-2000 problems began with popping hold-down bolt heads in the rear cam outboard drive gear. Engineers resolved this by not threading the bolt shank right up to the head, instituting a new tightening procedure and hardening plus grinding flat the washer underneath the bolt head.

Simultaneously, the same rear cam outboard drive gear's integral cast-in key began to shear in the rear-cam-journal

keyway slot. Andrews Products designed and made an aftermarket metallurgical superior rear cam outboard drive gear with a separate woodruff key that continues to work very well to this day on the first production Twin Cams. The Motor Company engineers successfully designed a new multi-spline male rear cam journal end mated with a similarly splined female attachment hole in the drive gear.

I am sure that both problems were a result of bean-counter cost-saving measures. I cannot visualize the engineers that I met at the Factory in March 1998, who taught me about the new Twin Cam engine, making such petty mistakes.

A couple of pinion shaft bearing irritations also occurred. A huge pinion bearing supports the right-side flywheel assembly in the right-half engine casing, as its pinion shaft protrudes into the timing chest from the bottom-end

compartment. The first nuisance was a loose fit between pinion shaft race and the inner pinion-bearing race. It was very difficult to isolate, but once done, the engineers tightened the tolerances. Then displacing air generated by descending pistons, traveling from the bottom end into the timing chest compartment between the individual pinion rollers and their mating cages made an annoying whistling sound. Of course, this was also offensive to the noise police with decibel-measuring machines at the ready. Tightened tolerances between rollers and cages resolved this situation in short order.

A major and destructive disintegrating rear cam outboard journal ball bearing caused much angst in 1999 and early 2000. Ball bearings are quieter than roller bearings, hence their use. The ball bearing was incapable of handling sideways stress from the offset primary outboard timing chain. A rear cam outboard journal roller bearing and slightly redesigned cam support plate put this destructive situation to

rest.

Other isolated problems occasionally arose. Some like stepped cam lobe surfaces were due, in my opinion, to accelerated parts production beyond machining capabilities.

While the following was never a burning public issue, I, as a mechanic, am very nervous about the use of INA early 1999-2006 SCE148 and the 2007 to present SCE168 rear journal cam bearings. The later bearings increase in inner diameter size a full .125” to 1.000”, which is surely an improvement. However, addressing the potential problem is through brute size rather than a different design bearing like the Torrington. I find advanced wear on these INA bearings frequently. At Heavy Duty Cycles (www.heavydutycycles.com), along with S&S Cycle (www.sscycle.com) and Crane Cams (www.cranecams.com),

we advocate the replacement of these particular INA bearings with their Torrington counterparts: B148 in place of the SCE148 and B168 in place of the SCE168.

I hope that the major problem occurring in the 1999-2006 timing chest design has been resolved by the 2006 Dyna and all 2007 and later Twin Cam use of hydraulic timing chain adjusters and rounded-edge chains. The early sharp-edged silent timing chains in combination with bouncy spring-loaded timing chain adjusters result in an adjuster pad prone to rapid wear, cracking, and breakage. Polluting debris from the pad is an ongoing concern to me. Simply, we need to get rid of this system for these and performance reasons that I am going to discuss.

However, a new problem may arise with Moment Arm torsional forces on the 2006 Dyna and all 2007 and later Twin Cam support plate parent-material bearings.

The 2006 Dyna and all TC96 cams are now five-piece so that appropriate metallurgy collaborates with journal ends and cam lobes. The Factory hardens them to differing tolerances specific to their usage and stress.

The oil pump has improved feed and much improved scavenging capabilities to help cure another unrelated problem in some Twin Cams and that is oil blowby also known as oil carryover or oil

All of these changes look good, but so did the original design in the beginning. Time will tell if these changes are successful although I will say that this system is an improvement.

In my opinion, performance applications require changing

these timing components to a dual-cam gear drive with the elimination of adjusters, timing chains, and sprockets in favor of ultra-reliable meshing gears.

## Higher Lift, Longer Duration Cams Need Stronger Valve Springs

Higher-lift cams need performance valve springs with sufficient travel to avoid coil bind of the spring coils. Heavier seat pressure valve springs ensure the valvetrain follows the dictates of wilder cam lobe profiles. This is especially necessary when the egg-shaped cam lobe rotates to a lower position to allow the valves to close, returning to their respective valve seats.

The Twin Cam timing chains and their mating sprockets have as much as or cam timing variance in the stock application. Initial cam timing variance with stock valve

springs is due to a relatively sloppy fit between chain and sprockets.

Performance applications should not tolerate this. Heavier seat valve springs can influence the weaker spring-loaded cam chain tensioners in 1999-2006 Twin Cams. Spintron dynamometer tests document an increase of additional cam timing variance as a result. The performance mechanic cannot abide this in the quest for a dialed-in engine seeking maximum power from its internal components.

Cam timing variance has a lesser influence with the consistent pressures of the new hydraulically actuated chain adjusters in the 2006 Dynas and all 2007 and later Twin Cams. My opinion is that these adjusters are similar to the components in the Shovelhead hydraulic lifters that work well with clean oil but are not up to the task of dealing with the

rigors of heavier seat valve springs over time.

There are those that disagree with this -- feeling it not of major importance -- but I have difficulty understanding lack of perfection when improvement is there for the taking.

Gear-driven cams have zero tolerance for valve springs attempting to sway cam timing.

Next, I will examine an impediment to allowing the Twin Cam to reach its performance potential, which is the head's capacity to flow air.

## Twin Cam Performance Obstruction

For emissions control reasons, the Twin Cam head ports do not flow as much air and gas as the Evo heads. We have a better design but less capacity for air-fuel delivery.

The intake valves are the same in both Evo and Twin Cam heads up until 2004 but the Evo exhaust valves have a slightly wider diameter head. At Heavy Duty Cycles, we routinely put Evo exhaust valves into 1999-circa 2004 Twin Cam heads to increase port flow.

The formula for power is simple. Gasoline will not burn by itself. It must mix in correct proportions with air to burn. The more burning achieved, the more expanding

heat developed, which translates into the increased pressure required to drive the pistons down on their power strokes. This creates more torque that gets the beast moving off the line and horsepower that keeps it going faster at speed. Therefore, the quest becomes breathing more air since delivering gas, especially with fuel injection, is easy and

precise.

The Twin Cam TC88 bathtub chamber is small having only 85cc volume, which is why the compression ratio is higher at between 8.8 and 9:1, compared with 7.5:1 for the Evo. Harley-Davidson raises this to a healthy 9.2:1 with TC96 models and a paltry 9.3:1 on the CVO 110-cubic-inch engines. This is paltry, in my opinion, because an engine this size will need more compression.

There are built-in torque valves more correctly called anti-reversion (AR) valves, one in each TC88 exhaust port. The term valve is not accurately descriptive because it is a step in the otherwise round exhaust port to help returning exhaust pulse gases bounce back the way we want, out the pipe. The TC96 heads eliminate the anti-reversion valve, depending on timing the returning exhaust pulses to bounce back off a closed exhaust valve.

Hot exhaust pulses leaving the combustion chamber travel down the pipe and into the muffler. They then reverse direction and pulse back up the pipe, completing this process many times. For the physics minded, these returning pulses explain by Kinetic Wave Theory and have much to do with temperature differences and resultant expanding and contracting gases and their pressure differentials. One of the functions of the muffler is to time these returning exhaust gas pulses so that they bounce off a closed exhaust valve, preventing them from reentering the combustion chamber, polluting the virgin intake charge.

The 2007 head has improved flow over the 1999-2006 varieties, although they base on the same casting and design. However, the flow is still not enough to service the larger engine from a performance perspective, as the increases are

marginal relative to what is possible.

Putting TC96 heads onto a TC88 engine will result in increased flow. Porting and polishing TC96 heads will give even more power to the anemic TC88. I say anemic because there is potential for more power in either engine. However, if we are going to the expense of supplanting stock heads, our dollar will be better spent on superior-flowing aftermarket heads.

Porting and polishing is always a good idea with either the TC88 or the TC96 heads.

Conversely, the Evolution V2's 80-cubic-inch 1340cc engine has heads that flow more air and gas than the Twin Cam 88 cubic inch 1450cc engine. Therefore, the 1340cc V2 heads also flow more air/gas than the upgraded 95-cubic-inch

1550cc engine.

The slightly improved flow of the 2007 heads must cope with servicing a larger displacement engine, so we return to square one. We now have a 96-cubic-inch engine whose main performance option will be to bore out its cylinders to produce 103 cubes.

I would not dream of building a 103-cubic-inch engine without much higher flowing heads.

Porting and polishing stock heads is very necessary in stock and mild performance applications. However, for anything more, we run into a second problem, as the valves are too small. Replacing stock valves with slightly bigger ones is easy for the performance mechanic and cost effective for the consumer. On the other hand, using the much larger valves found in aftermarket heads, while possible, may

become prohibitively expensive.

At what point does it become more economically feasible to upgrade to different heads altogether?

After analyzing certain aspects of the engine, it may seem overly deficient, so I will conclude with stating once again that the Twin Cam is the best platform ever designed by the Factory for extensive and reliable performance increases. I will give the various specific remedies and detailed cures for anemic head flow in Chapter VIII: Eliminating the Performance Obstruction.

## Stoichiometric Air-fuel Ratio Tendonitis

A new strain is increasingly beginning to aggravate the smooth operation of the newer Twin Cams, generated by its

old nemesis, EPA regulations concerning burning higher percentages of pollutants in the combustion chamber.

This aggravating and painful tendonitis has as its cause excess heat and the symptom of potentially harmful detonation. This results in actual harm in the new CVO 110s because the excess heat derives from ratios exceeding the Stoichiometric air-fuel ratio (AFR) for gasoline of 14.7:1 (14.7 parts air to one part gasoline). This is a very lean burning, and, therefore, heat-generating air-fuel mixture.

The reader may have run across the terms open loop and closed loop electronic fuel injection. I can tell you that I was very confused by this terminology until learning the very simple explanations. Closed-loop EFI, which we have on the newer Twin Cams, Sportsters, and V-Rods, means that during hot idle and cruising modes the air-fuel ratio is Stoichiometric 14.7:1.

Open loop EFI occurs at cold idle and during acceleration, when the AFR necessarily must be richened, adding more gas. Idle needs a richer gas ratio to allow the bike to warm up while acceleration demands more gas for power. Open loop with an AFR of less than 14.7:1 is wasteful in terms of emissions, while a closed-loop AFR ratio of 14.7:1 is theoretically the best for burning all the emissions.

The Stoichiometric ratio defines as the optimum air-fuel ratio (AFR), where all the available oxygen in the air combines with all the available fuel for burning in the combustion chamber. Maximum horsepower and torque will develop with an AFR less than 14.7:1. Too much lower and the opposite problem will occur. There will not be enough air to combine with all the gasoline molecules necessary for ignition and burning.

The cure for the detonation, heat-generated tendonitis is a lower AFR ratio. Gas is also a cooling agent as more gas lessens heat.

The CVO 110 is plagued with rear cylinder AFR ratios of from 14.8:1 to 16.0:1. To eliminate the tendonitis and gain much more power, I will lower the AFR to as low as 14.2:1. Please see Chapter III: H-D 110-Cubic-Inch CVO, subsection, Solving the Tendonitis of Lean Burning Air).

Many mechanics think the liner is moving in the cylinder, causing the rear head gasket to blow outwards from the top-end oil-drain hole. In Chapter III, subsection The Stoichiometric Fix: Aftermarket Cylinders, I discuss replacing the stock CVO 110 cylinders whose linings have great difficulty in coping with Stoichiometric and above heat…or do they?

1 Davidson, Bill. Harley-Davidson Archives, Juneau Ave. Milwaukee, Wisconsin. Speech, March 1998

2 Harley-Davidson. Service Bulletin TT352. Twin Cam Crankshaft Assembly Run-Out December 3, 2007.

# Chapter II

## Repairing the Achilles' Heel

Damage in the Achilles's heel, which thus far is the timing chest of the Twin Cam, evolves from EPA-mandated noise-reduction efforts.

### The Problem: Ovaling of Bearing Holes?

The current problems confine to the 1999-2006 cam timing chain apparatus.

However, I predict a potential difficulty may be developing

in the 2006 Dyna and all 2007 to present new cam-support-plate parent-material bearings. I will examine this situation in detail later in the section, Moment Arm Torsional Bearing Wear. This section will examine whether the parent-material design can override a law of physics.

Perhaps this law of physics will determine the component's planned obsolescence.

Evo's and predecessor engines' use of gears to actuate and drive the four valvetrains, each beginning with an individual cam lobe and ending with its mating valve, make too much clatter and whine for the noise police.

I have documented this subject at length in Volume I: The Twin Cam: Chapter II: 1999-2000 Teething Problems and Chapter III: Timing Chest Breakdowns and

First, I will briefly list the major teething and ongoing problems with the 1999-2006 timing chest designs.

Potential Ovalling is an Old Problem on a new Bike

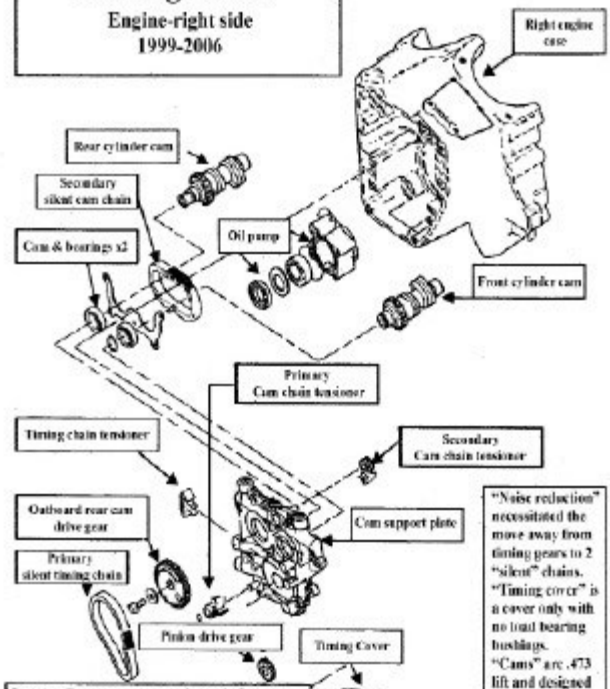
Next, I will use a law of physics, Moment Arm Force, to demonstrate the potential 2006 Dyna and all 2007 to present timing chest problems in the making.

Interestingly, the harbinger for this current and future improved timing chest setback occurred with loosely fit cam support plate bearings in the early Twin Cam years. Those of us who have done extensive timing chest work over the years sometimes noticed galling on the surface of the cam journal bearing support holes but didn't think much of it except to replace the cam support plate if bearing fitment was not a press fit. In a small percentage of cases, the galling

progressed. These galling increases the bearing hole size, which in extreme cases causes bearing ovaling of the support plate holes, particularly the rear one.

# Timing Chest

Engine-right side
1999-2006




Gerotor oil pump uses rotors instead of gears producing high volume and pressure. The gear driven oil pump used on Evos produces moderate volume and pressure. Gerotors fit over pinion shaft and are driven by it with the oil pump housing sitting over them. Oil pump is not visible as with the Evo. A new area oil is directed to by the cam support plate is to 2 piston cooling jets that squirt oil to the piston undersides. This relieves 50 degrees F from piston crown temp. Pump is dual scavange, eliminating the breather gear. Oil is picked up from both the crankcase and the timing chest and returned to the oil tank.

"Noise reduction" necessitated the move away from timing gears to 2 "silent" chains. "Timing cover" is a cover only with no final bearing bushings. "Cams" are .473 lift and designed to increase bottom end torque below 4000 rpm. An added benefit with 2 cams is straighter angled pushrods, which eliminates the Evo front exhaust "tick". Cam end bearings are beefier than Evo.

## Parent-Material Bearing Problem in the New Timing Chest Design

With the 2006 Dyna as the guinea pig, Harley-Davidson eliminated the use of more traditional ball and roller bearings to support the outer cam journals in the cam support plate. Instead, the Factory is using parent-material bearings, which are simple but precise holes machine-bored into the cam support plate!

Cost effective, for sure, but will this be as reliable as the proven rear cam journal support roller bearing and front cam ball bearing supports? I think not, even though I wrote in Volume I: The Twin Cam: Chapter VIII: The 2007 TC96 Engine, subsection, New Improved Timing Chest? that successful use of parent-material bearings has occurred in the automotive industry for a long time.

However, cars are temperature consistent as a result water jacket liquid cooling while air-cooled engines are unpredictably hot. Secondly, Harley-Davidson uses different metals with different expansion rates in its application. Briefly, the alloy cam-support-plate expands faster and greater than the steel cam journals that fit into the parent-material bearing holes. At room temperatures, the fitment of cam journal to parent bearing is precise. Under operating temperatures, there is greater tolerance because of differing expansion rates.

Will not protective film-strength oil bleed out via gravity on engine shut down?

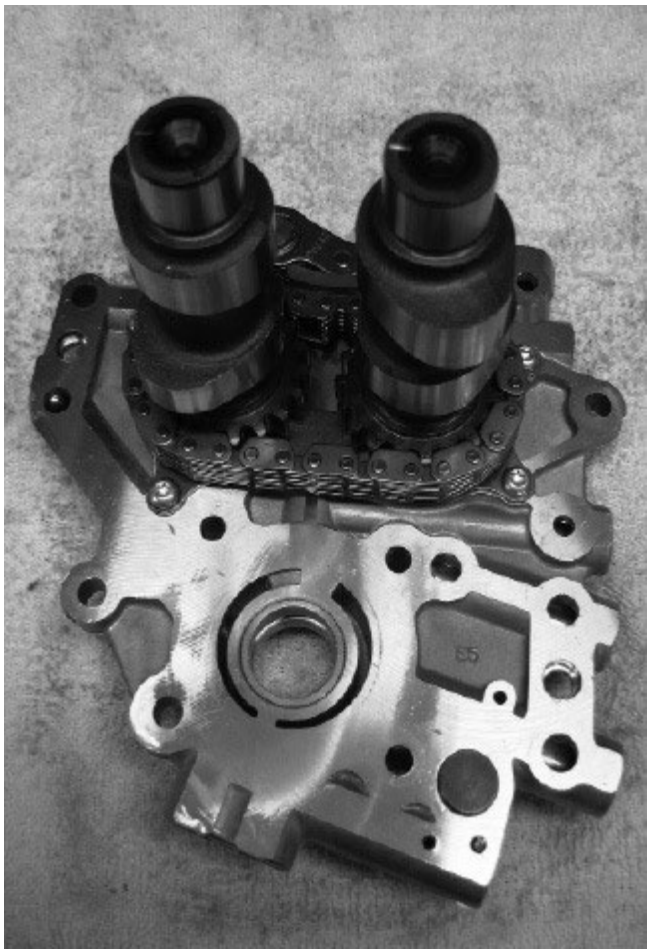
Will not the torsional stress of Moment Arm Force on start-up with a dry bearing cause the round bearing hole to oval

over time? How much time remains the question?

I have teased the reader with unexplained terminology and an inviolable physical law that I will detail in its simplicity later in this subsection, Moment Arm Torsional Bearing Wear.

Now let us look at performance problems and engine inefficiency in the use of timing chains and sprockets.

Inboard Cam Support Plate

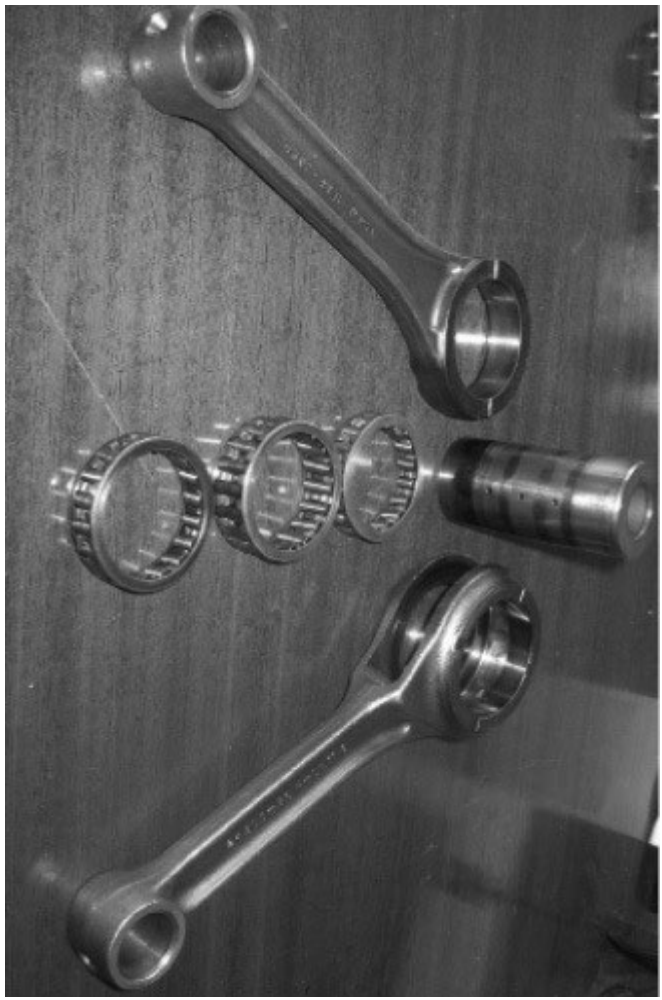


Pictured is the inboard side of a 1999-2006-cam support plate with cams. The secondary cam timing chain connects the two cams via their sprockets. Adjusting the chain tension is a spring-loaded cam chain adjuster, visible between the cams above the upper portion of the chain. The silent timing chains have relatively sharp edges. Loading and unloading valvetrain forces cause the timing chains to whip against the spring-loaded adjuster pads to prematurely wear, crack, and eventually break them. The polluting worn material enters the oil supply. Fortunately, the 1999-2006 10-micron oil filter and the later 5-micron filter eliminate most particulate. The pinion shaft bushing below the cams delivers oil fed by the internal oil pump into the pinion shaft (not shown) for delivery to the bottom-end crankpin, bearings, and parent-material big-end connecting rod races. Surrounding the pinion shaft bushing are the two half-moon linear oil feeds. The internal geroter oil pump sits on its partially supporting pinion shaft. The oil pump bolts to the inboard side of the cam support plate via the four holes surrounding the pinion shaft bushing. The

inside feed-and-return geroter inner diameters are slotted and fit a mating slot on the otherwise round pinion shaft. The inside geroters are rotated via the right-side flywheel pinion shaft (not shown) that protrudes from the bottom-end compartment into the timing chest. The pinion shaft extends just past the pinion bushing where the slotted pinion drive sprocket bolts onto the end of the mating slotted pinion shaft to power the outboard-side primary timing chain (not shown) that connects the pinion drive sprocket and the rear cam drive sprocket. Volume I: The Twin Cam has oil pump pictures in Chapter I: March 1998: The Introduction of the Twin at the end of Chapter Harley Oil Leaks: Marking Its and in Chapter VIII: The 2007: TC96 Engine. Flywheel, pinion shaft, crankpin, connecting rod and related pictures may also be viewed in Volume I: The Twin Cam in Chapter I: March 1998: The Introduction of the Twin Cam, Chapter II: 1999-2000 Teething Problems and Chapter IV: Harley Oil Leaks: Marking Its Spot. Cam support plates, cam drive sprockets and timing chains may also be examined in Volume I: The Twin Cam in

Chapter I: March 1998: The Introduction of the Twin Chapter 1999-2000 Teething Problems and Chapter VIII: The 2007: TC96 Engine. I do not give the current page numbers, as they will change each year as I update this first volume with changes to the engine and more understanding on my part evolves.

Disassembled Twin Connecting Rod Set



This picture may look out of context on first blush. I will compare the connecting rod parent-material bearing races with those found in the late-style cam support plates. First, I will describe the component parts in this set of 1999-2006 Twin Cam connecting rods. The 2007 to present TC96 connecting rods are the same except the small ends that house the wristpin bushing do not square off as in this picture, but narrow to the top. This lessens reciprocating weight to offset the effects of the increased stroke from TC88 4.000" to the TC96 4 ⅜". The small ends have a hole in the top of each con rod for gravity-drip-feeding oil to lubricate the piston wristpin and mating wristpin bushing. The hollow crankpin is a first for Harley-Davidson. End caps plug either crankpin end to form the center oil reservoir, which has a cooling effect. The three oil holes individually pressure-feed the three-captured bearing sets that rotate between the crankpin and the parent-material big-end connecting rod bearing races. Pre-1999 con

rod big ends fit with separate rod races that pressed into the big-end holes. The slots on the sides of the con rod big ends are oil grooves for lubricating the male rod sides to the female rod insides. The outer female rod's big-end sides need lubrication via these slots as they rotate against the inner flywheel thrust washers. Splashed oil in the bottom end compartment also gravity-drips down onto and in between the big ends. The term captured means that the roller bearings are a slight finger press fit to pop the rollers into the roller cage. The roller cage keeps the cylindrical rollers in alignment with their rotating movement travel. They run square, prevented from any sideways drag, which will cause bearing skid. Interestingly, the oil mythologists purveying their ignorant mistruths blame bearing skid on synthetic oil being too slippery. The true cause of bearing skid is misalignment; the bearing cage wears to the point the bearings cock a little sideways. They can no longer roll and instead drag. If anything, synthetic oil will protect this destructive bearing for longer than fossil oil. Parent-material bearings are not a

traditional ball or roller bearing with actual rotating bearings. The con rod parent-material-bearings are the actual big-end holes in the rods that are machined smooth to size, hardened and remachined to gain precise fitment. One might think, incorrectly, that the Twin Cam con rods wear out because of the bluing from the hardening process. The three connecting rod parent bearings, (two in the female rod and one in the male rod), rotate around three sets of captured rod roller bearings that separate the big-end parent-material rod races from the crankpin. In this particular case, the parent-material bearings (rod races) are a very reliable setup. The roller bearings diffuse any torsional stress; there is a constant separate oil feed to each set of captured bearings and the mating components, including the crankpin, bearing sets, and con rod races (parent-material bearings). They are all hardened steel with the same expansion rates. This contrasts with the aluminum parent-material bearings in the late cam support plate that expands more and faster than the hardened steel cam journals, they support. Moment Arm torsional force,

differing expansion rates, and oil drainage as a result of greater alloy expansion may combine to oval the cam support plate parent-material holes. Flywheel, crankpin, connecting rod, and related pictures may be viewed in Volume I: The Twin Cam in Chapter I: March 1998: The Introduction of the Twin Cam, Chapter II: 1999-2000 Teething Problems and Chapter IV: Harley Oil Leaks: Marking its Spot.

We discovered two new phenomena early in 1999 when installing performance cams into the new Twin Cam engines. First, mechanics discovered that mild cam changes did not add much, if any, noticeable horsepower in the TC88's Secondly, sometimes when installing wilder high-performance cams and valve springs in the TC88, not all the performance expected was always forthcoming.

Forgive a small digression: if I jump forward to 2007, it is interesting to note that all the performance horsepower one would expect from the eight extra cubic inches in the TC96 are also not forthcoming, as one would expect, although for different reasons.

From a performance perspective, there is a big quandary with both the loose fit of the cam-drive sprockets that fit into the silent timing chains and the strength of the chain

tensioners. The loose fit allows cam timing to waiver by as much as 4-degrees. One only has to look at these sprockets and timing chain fitment to see that exact timing is impossible.

Sloppy part fitment leads to sloppy cam timing.

This certainly does not happen with a properly fitted spur gear drive, helical gear drive, serpentine belt drive or a carbon fiber belt drive.

Furthermore, performance valve springs will dominate the weaker adjuster tensioner springs in 1999-2006 timing chests and may challenge over time the hydraulic adjusters in the later timing chests. As a result, cam timing can be off as much as additional 5-degrees. Performance springs have a higher seat pressure in order to ensure the valves follow the dictates

of the cam lobes thus putting more pressure on the timing chain adjusters.

Performance cams with higher lifts and different lobe profiles require stronger seat pressure springs with more room between coils. The stronger and spaced-coil valve springs become the force to alter increased inertia and momentum generated by performance cam-lobe-profiles.

There is no predictability as the constantly oscillating cam timing may near the designed coordinates, move into advance, or retard positioning. The result is that the spark plug may not always ignite the fuel at the optimally designed ignition timing. Therefore, the pistons may not be at the most advantageous compression and power stroke to deliver engine efficiency and its compatriot, performance.

Ask the hard questions!

Namely, how do we know about this cam-timing variance? In addition, how do we know these figures of an initial stock engine and up to an additional performanced engine, are accurate? Well, S&S Cycle has high-speed videos that document that these cam-timing variances from loading and unloading of the timing chains on their adjusters in fact do occur.

Scott Sjovall, who is in research and development and a brilliant technician at S&S says, “There is a horrific amount of harmonics in the V-twin valvetrain. There are several causes of these harmonics. One occurs when the tappets go over the nose of the cam and create a reversing load. Since we are dealing with only two tappets on each camshaft in the Twin Cam, this is a significant problem compared to multi-cylinder cams. They tend to have multiple lobes that offset each other.

Other causes are pushrod and rocker arm flex and spring surge. All of these problems are aggravated when you elevate valve spring forces, especially in a Twin Cam engine. The different harmonics affect the chains and tensioners. The proof for all of these claims can be seen with our high-speed videos of the chain tensioner movements under different spring loads." 1

A real nifty tool called a Spintron Dynamometer analyzes valvetrain dynamics in its S&S engines. The Spintron is an engine stand that uses compressed air to spin the engine to the desired rpm. A high-speed video camera, sometimes used in conjunction with a strobe, records the action of whatever the technician is interested in, be it valve springs, pushrods, cam chains, or whatever. This cool tool gives live on-time results on a screen plus document print analysis for later use.

This unique dynamometer will also do durability tests and component failure analysis, and document frictional horsepower loss for interacting parts, such as the valvetrain. A laser tracking of valve bounce, valve floating, adverse valve spring harmonics, and deflecting pushrods document these characteristics to demonstrate that parts are working properly or need more development. The monitoring of these components from one specified rpm to the next and onto the next is crucial since a part may work fine at one speed but not at another.

Spintron cycle testing simulates hundreds or thousands of race or street miles on your engine.

Thanks to the Spintron, we now have proof that the Twin Cam timing chains do indeed load and unload, causing unwanted fluctuations in cam timing through both live video and print documentation. Unwanted cam-timing variance directly affects horsepower gain achievements with

performance cams and, to a lesser extent, with stock cams or mild bolt-in ones.

Full-Contact Quiet Helical Gears

An alternative choice from timing chains was a helical gear drive to replace the spur gears clickety-clackety when cold and whining when hot.

The always-in-contact helical gears are quiet.

Every potential solution brings its own engineering dilemma to resolve. In the case of helical gears, the quandary is thrusting forces along the axis of the angled-gear teeth.

Serpentine belts will probably be the final solution if Harley-Davidson can manage to keep their beloved pushrod-perated overhead (OHV) valvetrains in lieu of the more

efficient, less-marketable overhead cams (OHC).

## The S&S Three-Cam X-Wedge Engine

S&S Cycle is ahead of its game as usual in meeting present and future government regulations by developing the X-Wedge, its engine of the future. The X-Wedge is an air-cooled V-twin with pushrod-operated overhead valves that meet the 2008 CARB (California Air Resources Board) and 2010 EPA regulations. I will be looking at the X-Wedge in detail in Chapter XIV: The subsection, The S&S X-Wedge.

One major advantage of the H-D Twin Cam is that the dual cams substantially lessen but not eliminate the severe pushrod-lifter angles that occur in previous H-D engines.

The X-Wedge carries this theme a step further. S&S has

almost eliminated any pushrod-lifter angles in favor of the much more efficient straight up and down configuration. This new engine uses three The center one services the two intakes, which have always been the straightest up and down pushrods on any of the engines discussed here. There is a separate exhaust cam on either side of the dual intake cam.

Three serpentine belts drive the large-base circle cams, thus eliminating gears, chains, and sprockets. A flat rubber drive belt turns multiple pulleys in the timing chest.

The serpentine belt gains its descriptive name by snaking around the various pulleys. A spring-loaded pulley maintains tension on the serpentine belt, doing away with the need to retension replacement belts. Serpentine belts generally last 25% to 50% longer than conventional V-belts.

The X-Wedge has many other novel features that eliminate traditional boondoggles and resolve government policies planned for the immediate future.

## Gear Drives

S&S Cycle was the first to develop gear drives to resolve timing chain breakdowns and to offer reliable and accurate cam timing. Now many companies offer the camshafts themselves with their own cam-lobe-profile variations. However, most still use the two S&S inboard cam-support-plate cam gears, and the outboard pinion drive gear and its mating rear cam drive gear.

S&S warned me that a gear drive would not be as quiet as the stock timing chain setup because Harley used loose tolerances with the support plate. The cam support plate dictates the cam locations and thus the gear lash. The first

bike, a 1999 Road King, we did an install on was with an S&S 585G gear drive, which belied the noise warning. It ran quieter with the S&S gears than it did with the timing chains.

I have never been impressed with the Twin Cam's supposed quietness. The prototypes that I viewed in Milwaukee were very quiet, but my own Road Glide at the time was as noisy as properly installed Evolution timing gears.

After installing hundreds of S&S Cycle, Andrews, Crane, and Wood Performance gear drives in the last nine years, I feel the truth of the matter is that some gear drives are noisier than timing chains, some emit the same noise decibels, and others run quieter.

Production tolerances in mass manufacturing are necessarily looser and vary more than handbuilt prototypes.

Pictures of stock timing chain apparatus and S&S gear drives may be examined in Volume I: The Twin Cam in Chapter II: 1999-2000 Teething Problems, Chapter III: Timing Chest Breakdowns and Solutions, and in Chapter VIII: The 2007: TC96 Engine. I do not give the current page numbers, as they will change each year as I update this first volume with changes to the engine.

## Belt Gear Drives

Bob Wood of Wood Knight Prowler cams fame (www.woodcarbs.com), has already done this with a kit that he has developed using a carbon fiber belt to power the pinion, and rear-cam-gear drive pulleys replacing the sprockets and primary drive chain. This 5mm reinforced carbon-fiber belt connects the replacement drive pulleys powered by the rotating pinion shaft. I use the H-D

nomenclature for the sprocket/pulley for reference only as Bob's invention utilizes a 5mm cam gear with an adjustable spider gear.

Bob probably will not like me calling him an inventor since he is an unassuming guy. He feels he has just adapted automotive technology. Maybe so, but I think he is a pretty inventive person with a knowledge base that surpasses most in the Harley world.

The pièce de résistance for this timing chest fix is the use of quieter full-contact helical gears to drive the gear drive cams instead of noisier spur gears. I will be looking at Bob's Knight Prowler Advanced (Silent) Belt Drive System, in detail later in this chapter under the subsection: Knight Prowler Advanced Belt Drive System. You know, I can see Bob's ideas incorporated into future Harley design, especially if the

revamped timing chain system introduced with the 2006 Dyna becomes problematic over time.

Let us not even think about a water jacket enclosing our beloved engine with a big cooling radiator out front! All of this will be necessary at some point in the future for controlling noise as well as regulating engine heat. Innovators like Bob Wood may push off the inevitable.

## The S&S Solution

I have documented briefly earlier in this chapter and extensively in Volume I: The Twin Cam, Chapter III: Timing Chest Breakdowns and Solutions the downside of cams utilizing cam sprockets and timing chains, particularly when using performance valve springs, which may be stronger than the chain-adjuster springs.

I do not see how it is possible to run anything but gear-driven cam sets when heavy seat pressure valve springs are required for accurate cam timing. Furthermore, to develop a truly reliable and fast Twin Cam engine, headwork, compression increases, and bigger cubic inches all require coordinated modified cams and therefore the heavier valve springs.

Moreover, make no mistake; these springs are very necessary with the higher lifts. The higher the lift, the more valve-spring-seat-pressure is required. In my opinion, high lifts and matching springs will prematurely wear out the timing chain adjusting parts.

In addition, Spintron Dynamometer documented cam-altered timing may vary by as much as five more degrees. What else do we need to know?

Colliding Camshaft Lobes

The mating cam lobes on the two respective rotating cams come close to each other for an instant every 360-degree.

Cam lobes from each of the cams can interfere with each other and collide with higher lifts. S&S first resolved this engineering problem on its high-lift gear drive cam applications. Interference between cam lobes is not an issue because S&S reverses the rotational direction of the rear cam. With the cam lobe reversal, the valves still open and close as originally designed. I will discuss at length how gear drives and timing chain drives resolve this issue in Chapter IV, Cams, Cams, Cams.

I have briefly examined increasing engine efficiency by tightening up cam timing specifications to ensure that the

spark plugs ignite the fuel mixture when the pistons are at their optimal positions. The replacement of the timing chain apparatus in the Twin Cam with a precise aftermarket gear drive prevents cam-timing wandering.

One thing I have not mentioned is emissions control parts, which are the exhaust, the ignition, fuel delivery ratios, and cams. The stock cams have restricted lobe profiles designed to burn the most fuel possible to reduce harmful emissions. This sounds good because burning all the fuel also creates power. However, the cam designers do this without taking advantage of physical laws of nature such as inertia, momentum, and scavenging by opening and closing valves near the theoretical of crankshaft rotation allowed each cycle of the four engine strokes. There is little valve overlap and minimal cam duration for valve opening time. Increased lift, valve opening and closing timing, intake and exhaust valve overlap and longer duration make power, but stock cam profiles are

deficient in these areas. Many times these stock lobe profiles suppress performance or power that is already there in the overriding interest of reducing emissions in naturally aspirated engines. However, the stock cams work wonderfully with forced induction; turbocharging, supercharging and nitrous oxide. Please see Chapter XIII: Forced Induction for the compelling reasons.

The above factors, in conjunction with the engine-management system and fuel delivery combine to continually increase the ratio of more air relative to the fuel. The engine management syatem consists of sensors feeding information to the electronic control module (ECM also known as an ECU or ICM), ignition. One of their functions is to create heat to reduce emissions. Increasing proportionate air with less fuel creates heat. The excess heat burns a higher percentage of the less fuel ingested.

Again, this may sound contradictory: more fuel burning but with less resultant power. A higher percentage of burn of a smaller amount of fuel is very efficient. However, there is still less fuel in the combustion chamber combining with a higher percentage of air.

There is one further reason for replacing the stock timing chain apparatus to a gear or a belt-gear drive, as it will benefit the Twin Cam over time with reliability and longevity.

## The Trick: Meet Emissions Dictates with More Power

In fairness, Harley-Davidson must adhere to government regulations as difficult as they may be for a two-cylinder, air-cooled pushrod-operated engine.

We cheat the system to gain our much-desired performance gains by removing or modifying emissions

controls. Authorities have overlooked and in some cases tolerated these indiscretions for decades. The window is closing rapidly. Abuses from the loud-pipe crowd have accelerated the process. Shaking the windows of hospitals and schools with noise is not conducive to ongoing tolerance.

I love the sound of straight pipes, but I, like many thinking riders, learn to feather the throttle in situations that will offend others. It saves on costly ticket infractions also. It is ironic that in most cases, straight pipes cause a loss of power and especially low-end torque.

The future is developing performance options that also meet government mandates.

Some think this not possible. I am not one of them.

Performance pipes have always made less noise than straight pipes. These exhausts are also quietening further with progressive companies applying technology to the noise reduction issue.

EFI allows for potentially huge power gains. Precise fuel delivery on demand both helps reduce emissions and increase power.

The challenge is to meet noise and exhaust emissions at the same time as increasing horsepower and torque. Bob Wood's innovative belt-helical cam gear drive does exactly this. This is all possible. Emissions controls do not have to equate with power decreases.

The solutions are new ideas, adapting proven automotive know-how, and developing new technology.

The next section looks at the H-D Solution.

While the jury is still out on the parent-material bearings, the hydraulic adjusters will enforce stricter cam timing. A simple Factory change from spring-loaded timing chain adjusters to more consistent hydraulic ones to resolve adjuster pad breakdown have at the same time given a performance improvement of better cam timing accuracy, albeit not as good as a gear drive but a step in the right direction nonetheless.

A later section, The Andrews Solution, takes the 2006 Dyna and all 2007 to present Twin Cam timing chest design and retrofits it to the 1999-2006 Twin Cams to eliminate its inherent timing chain-adjuster pad wear problems.

## The H-D Solution

As I have related before, Harley-Davidson changed the type of chain and tensioner on the 2006 Dyna and all 2007 to present Big Twins. The new chain has a rounded edge, the tensioner pad is from a different material (nylon), and the tensioner is precisely and consistently hydraulically actuated and not susceptible to the vagaries of spring-loaded tensioners.

The rear cam journal INA SCE148 (OEM 9198) support bearings in 1999 to 2006 Harley-Davidson's judgment are not a problem. I differ with the Factory perception as we have just taken too many Twin Cams apart for performance reasons, only to find what we consider adverse wear on these bearings. Furthermore, many of these engines had minimal mileage. As I have said, S&S Cycle and Crane Cams supply replacement Torrington bearings with their cam-gear-drive kits to replace stock INA bearings. At Heavy Duty Cycles, we replace the stock INA SCE148 (OEM 9198) bearings with Torrington

B148s in our entire Twin Cam engine work in the 1999-2006 models.

Harley engineers made changes to the INA bearings in the 2006 Dyna and all 2007 and later Big Twins. H-D stubbornly stays loyal to the INA bearing but increases its size from the 1999-2006 .875” a full .125” to a new and bigger 1.000” inner diameter. This, of course, means that H-D has also increased the rear cam journals to a 1.000” outer diameter. The bearing is definitely stronger than what went before, perhaps a design admission by the Factory that it, too, must have seen adverse wear on the smaller INA bearings used from 1999 to 2006. Harley finally made these and other dramatic changes on one tester model in 2006.

The 2006 Dynas attempted to address 1999-2006 timing chain problems with a new system replacing the silent cam

chains with a roller chain, a redesigned cam support plate, new hydraulic cam chain tensioners, and increased load capacity cam bearings in the engine casing. All the 2006 Softail and touring Big Twins retained the earlier 1999-2006 timing chest mechanism. In my opinion, the 2006 Dyna was a guinea pig for future change.

These modifications carried over into all the 2007 to present Big Twins. Some changes seem to be very much better and will eliminate some of problems in the past.

The new roller chains do not have sharp edges like the 1999-2006 silent cam chains.

Hydraulic units on the 2006 Dyna and all 2007 to present Big Twins have replaced the troublesome and unpredictable 1999-2006 spring-loaded tensioners. They work much the same as a hydraulic lifter with a small spring inside to help

tension while the hydraulic pumps up. An even and consistent amount of tension produces. This is very similar to the hydraulic tensioners that have successfully adjusted chain slack with the counterbalance chains in the Twin Cam B models since 1999.

Nylon replaces the old-shoe material.

In 2006, some engineers finally began acknowledging in private that the 1999-2006 tensioners often suffer excessive wear. Only time will tell if changes will resolve previous wear issues. Some changes like the rounded timing chains and the hydraulic nylon adjusters look to be real improvements.

Often when a manufacturer like H-D makes dramatic changes to fix one problem, new ones may evolve from the changes. One major concern I have with this new system is

the elimination of cam-support-plate cam-journal-bearings in favor of parent-material bearings. Parent-material bearings are just what the name says. The parent-material of the cam support plate is the new bearing. I will look at this more closely in a following section, Moment Arm Torsional Bearing Wear.

## The Oil Pump

The new oil pump (26037-06) fits 2006 Dynas and all 2007 to present Twin Cams. It is superior to what went before; it has a different porting arrangement than earlier ones.

This new pump will not mate correctly with 1999-2006 cam support plates, except the 2006 Dyna. The original 1999 oil pump (26035-99) has gone through various improvements (26035-99B) through the years. However, the 26037-06 oil

pump, pumps 10% more oil on the feed geroter side. Geroters are lobed gears that run inside each other.

Perhaps its biggest benefit will be in attempting to control oil carryover also known as blowby or external venting, where an oily mist and sometimes straight oil expels from the top-end breathers into the air cleaner and out onto the side of the engine. The 2006 Dyna and 2007 to present Big Twin oil pump has 23% improved scavenging capacity.

Pre-1999 Big Twins always used mating spur-type gears to pressurize the oil via their close proximity to the inside oil pump body cavities. Gear pumps produce moderate pressure and moderate volume.

A geroter system is the best style pump in use for Big Twins, Twin Cams, and Sportsters. Geroters produce higher

pressures and higher volumes of oil over the gear-drive oil pumps. The geroter pairs are narrow and flat. One rotor sits within the other with a total circumference of about 1 ¾". The outside rotor is round on its outside perimeter. The hollowed-out center has internal nubs much like a belt drive belt. The outside perimeter has external nubs that interact with those on the inside of the outer rotor. The mating nubs move in a seemingly eclectic but steady pattern because the inner geroter has one less nub or lobe than the outer. The meshing lobes open to let in oil and close in on each other, squeezing trapped oil out to make pressure.

The 1999-2006 (except the 2006 Dyna), oil pump has a twelve/thirteen lobe interaction. The outer geroter has thirteen internal lobes while the inner geroter has twelve external lobes. Therefore, with the new 2006 Dyna and all 2007 to present Twin Cams ten/eleven-nub rotors, more oil feeds into the engine. More oil also removes from the timing

chest and returns to the oil tank instead of pushing into the top end by the voluminous air displaced in the engine.

This air movement occurs every time the pistons descend in their cylinder bores.

The oily air forces into the timing chest cavity and up through the pushrod tunnels, sometimes overwhelming the breather apparatus in each head. The breather apparatus design separates oil from air so that the air alone vents into the air breather. The vented air reingests into the engine as an emissions control.

Retrofit 2007 Oil Pump to 1999-2006 Engines

A new Screamin' Eagle (www.harley-davidson.com), product introduces a billet cam support plate, and a high-volume oil pump kit (25283-07), that fits 1999-2006 Touring

models and Softails, as well as 1999-2005 Dynas.

The oil pump has 8% greater capacity on the feed side and 22% more scavenging on the return side. The increase in oil feed pressure particularly to the piston cooling jets will help cool engine oil temperatures, or, more precisely, allow the oil to transfer more heat to the head and cylinder cooling fins. This also sounds like a great time to install a ten-row Jagg oil cooler or two and synthetic oil to replace the lower-ability fossil oil.

In addition, more oil will deliver to the cam tensioner chains. I hope that this will allay for a little while anyway any impending adverse wear in this timing chest area. Finally, the increase in oil to the pinion-bearing bearings and the bottom-end connecting rod bearings is always a good goal up to a point.

Pictures of early and late stock timing chain apparatus and gear drives may be viewed in Volume I: The Twin Cam in Chapter I: March 1998: The Introduction of the Twin Chapter II: 1999-2000 Teething Chapter III: Timing Chest Breakdowns and Solutions, and, in Chapter VIII: The 2007: TC96 Volume I: The Twin Cam also has oil pump pictures in Chapter I: March 1998: The Introduction of the Twin at the end of Chapter IV: Harley Oil Leaks: Marking its Spot and in Chapter VIII: The 2007: TC96

## The Andrews Solution

### Retrofitting 2007 Timing Components

John Andrews of Andrews Products (www.andrewsproducts.com), who is always on top of his game, has developed a kit combined with H-D parts to adapt

the 2006 Dyna-2007 to present Twin Cam roller chains to all predecessor TC88 engines.

The new Harley-Davidson timing chest components are for the most part superior to what went before, even after resolving the early 1999-2006 teething problems. Chain tensioner pad wear is the most annoying to this day. There is no predictable service life for these early spring-loaded adjuster pads. I have seen some last for forty thousand miles or more. Others damage with first use. We routinely replace them in Twin Cam engines after a few thousand miles. There is no warning, no noise!

Many times, we at Heavy Duty Cycles only find this defective part wear beause we are doing performance work. Of course, with our experience we now look for parts wear.

Checking drained engine oil thoroughly can give indications of wear debris. Cutting apart oil filters, expanding the pleated synthetic media, and examining it in the sunlight will show if there is evidence that the parts have worn. However, this may be too late, better late than at engine breakdown time.

Shoe wear reduces as the new roller chains have rounded edges. The hydraulic adjusters also give more even and commanding support.

The Andrews kit uses the late-style 2006 Dyna-2007 Harley-Davidson cam support plate (25355-06); the primary (25675-06) and secondary (25683-06) cam drive chains; the hydraulic chain adjusters (39968-06, 39969-06); the new oil pump (26037-06); the seventeen-tooth crankshaft cam drive sprocket (25673-06); the thirty-four-tooth cam drive sprocket (25728-06), and the camshaft chain and fastener kit (25585-06). The conversion also requires some other minor H-

D parts.

Andrews has manufactured performance cams that will adapt the above parts to the earlier 1999-2006 Twin Cam engines. The 2006-2007 new-style Andrews cams for 1999-2006 fitment use two new 2006 Dyna-2007 Big Twin-type camshafts except the camshaft inner bearing journal size is machined down from 1.000” to .875 “ in diameter in order to fit into the smaller inboard cam bearings INA SCE148 (OEM 9198) or the Torrington B148s that I prefer and use. Inner cam journal size is one reason that the new-style cams will not retrofit to earlier Twin Cams and vice versa.

Another difference is that the cam chain sprockets are thinner than on 1999 to 2006 Twin Cams. Except for the camshafts, all of the above parts are stock Harley-Davidson. We can now adapt the new roller chains to older TC88 motors

and get away from the silent chain systems, which have been so prone to rapid wear and early failure. Furthermore by using a 1999-2006 cam support plate and a 2005-type oil pump (26035-99B), it is possible to adapt gear drives to 2006 Dyna and later Twin Cam engines. However, the adaptation will require manufacturing cams with 1.000” inner bearing journals and smaller outboard journals to retrofit into the previous cam support bearings. Supporting the outer cam journals will be the reliable front ball bearing and rear cam roller bearing.

Gear-drive cam sets for 2006 Dyna and 2007 to present Big Twins without using 2005 components; require new design camshafts and drive gears, which became available from manufacturers like Andrews, Crane, S&S, and Wood in the latter part of 2007.

In conclusion, I have three comments.

The first is we do not want to give up the new oil pump with more 10% feed and 23% more scavenging capabilities.

Secondly, the hydraulic chain adjusters are an improvement we want to keep.

Thirdly, we will also have to wait and see if the new cam-support-plates with parent-material bearing holes will prove as reliable as the previous ball bearing-roller bearing setup.

Perhaps the next generation timing-chest components will have the 2007 to present bigger rear cam journals, the new hydraulic chain tensioners with the roller chains and the new oil pump combining with a cam support plate with a front ball bearing and a rear roller bearing like the late 2000 to 2006 plate. This will get rid of the good but less effective earlier oil

pump, the silent chains, spring-loaded tensioners, and the new parent-material support for the outboard cam journals.

## Moment Arm Torsional Bearing Wear

Many times, I am only a few steps ahead, as I, too, am constantly learning on a steep curve. Information may come from a reader via e-mail that has expertise way beyond mine in a particular field. I may be reading industry-expert interviews in the auto section of a local newspaper and learn tidbits about alternative fuels or new emissions dictates coming down the line. The next thing you know, I am off researching and expanding on the information byte and, a new article is born.

Through reading related mechanical stuff, many times I glean a formerly missing piece of information to a conundrum

in my mind. Oftentimes this seemingly innocuous piece of information may allow me to solve a mechanical problem, come to an understanding of complex engineering theory, or dissect a law of physics. Then it is a short analytical step to figure out how these principles apply to Harley-Davidson, many times the answer coming while I am sleeping.

The beauty is that my diagnostic dreaming reduces the complex to an easy understanding. Once I understand, it is relatively effortless to communicate my realizations to my readers. As oft happens, information arrives at my doorstep.

This time a fellow named Jeff Davies walks into my store, Heavy Duty Cycles in Toronto to buy my first book, Volume I: The Twin Cam. He is abuzz, bursting with enthusiastic encyclopedic knowledge he just could not contain. Originally, an industrial machinist Jeff became an inventor of and patent holder of automation equipment, proficient in CAD design, and a writer of automation trade manuals.

Moment Arm Torsional Force

Jeff began talking rapidly about something called a Moment Arm, which causes torsional imbalances affecting the new 2006 Dyna and all 2007, and later Twin Cam outer cam journals and the parent-material bearings in the aluminum cam support oval plates.

He said that aggravating this imbalance are the differing metal expansion rates and oil drainage from between the cam journals and the parent-material bearings when the engine is at rest and beginning to cool.

Whoa!

What is a Moment Arm and how do you spell that, buddy?

First Jeff qualified what he was about to explain: “In the industry I am in we design and manufacture synchronous machinery. The motorcycle engine is the same. Camshafts open/close valves in perfect timing along with many mechanically precise components. Any design, has a flaw(s), this is a fact. The challenge is to provide a design where the flaw has the least impact on the product supplied.” 2 Jeff then verbalized his thesis on why the parent-material bearing holes in the new cam support plates would not last. He said, “When analyzing any mechanical system that is experiencing problems, I think it is important to establish what forces are exerted on the applicable components and at the same time ask if the components are able to handle forces outside of their intended purpose.” 3

There are three parts to understand and apply to a parent-material bearing: Moment Arm torsional forces, uneven metal

expansion, and lubricant drainage.

First, let us analyze the immediate past.

## Late 2000 to 2006 Cam Support Plate Bearing Assessment

“A bearing [roller or ball)] is designed to handle a radial load. Any axial loads in combination with radial loads could be a problem…I think the [ball] bearing was coming apart because of the angular moment arm at the end of the shaft…The roller bearing would take this type of load better than a ball bearing because of the Length vs. Diameter Ratio of the roller. A double row of rollers would definitely handle this but I suspect this would be a huge redesign of the thickness of the support plate.” 4

Time has proven the superiority of the rear roller bearing

in handling both radial and axial loads from late 2000 to 2006 over the 1999 to early 2000 destructive rear cam journal ball bearings. The roller bearing is very reliable for this purpose. Its long cylindrical-shaped rollers contrast much with the small round balls found in the ball bearing, especially when visually observing the self-explanatory length-versus-diameter ratios of both.

The ball bearing has a ratio of 1:1 while the roller is close to 3:1. The rollers are able to withstand the rear cam axial loads that destroy the ball bearing in a relatively short time. However, Jeff was correct in his assessment that a double row of rollers would add more than double the protection and longevity. Furthermore, his redesign comments were right on the money.

Did I say “money”?

The redesign costs are prohibitive to value gained since the single roller bearing is doing an adequate job in late 2000 to 2006 Twin Cam support plates.

## Moment Arm Theory Applied to Ball and Roller Support Bearings

This Moment Arm Force law of physical theory also applies to the 1999-2006 cam support plates with bearings.

However, the correct bearings, roller in the rear and ball in the front, properly fitted withstand Moment Arm torsional forces because they are hardened steel with the same expansion rate as the cam journals. Furthermore, the individual rear roller or front ball bearings can turn. They rotate in their cups to absorb applied Moment Arm torsion. Thirdly, the entire inner bearing race and the complete set of

rollers or balls can turn with the torque that needs to dissipate its twisting energy. This is dependent on the outer race not being able to rotate in its support plate hole.

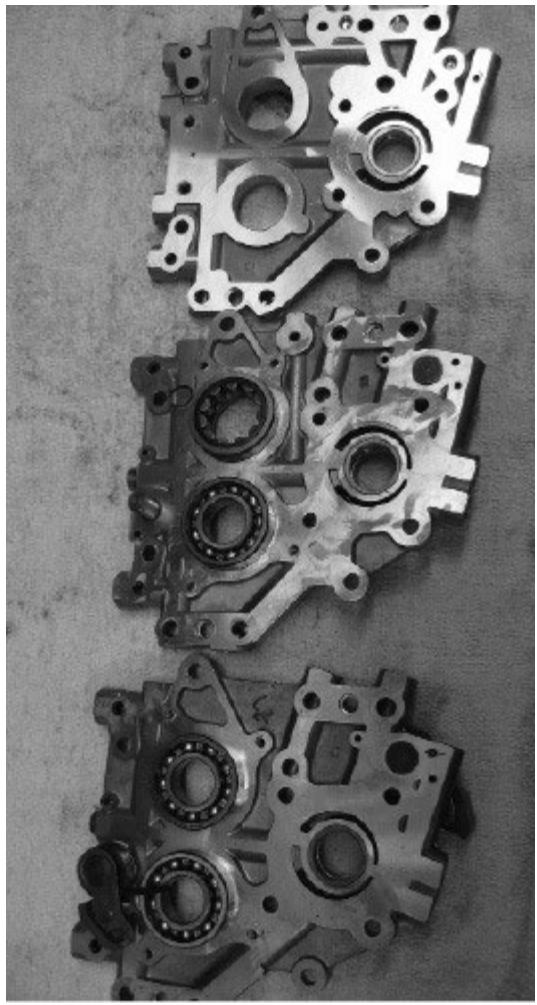
Parent-material bearings are softer than a hardened steel bearing, expand at a greater rate than the cam journals, and do not have roller or ball bearings to absorb the torsional forces, particularly on start-up.

Once understood, start-up of a cold engine, lack of sufficient lubricant separating cam journal and parent-material bearing, with the sudden torsional force of the rotating cam journal in the parent-material bearing…well, won't the softer aluminum round hole begin to prematurely wear?

The Moment Arm of torsional start-up force will wear a supporting bearing hole unevenly, eventually making it oval,

which is obviously not a good thing.

1999 to Present Cam Support Plates



Pictured are the three major cam-support-plate styles in use from 1999 to present. The top cam support plate had first use in the 2006 Dyna models and all 2007 to present Twin Cams. A preceding picture, Inboard Cam Support Plate, near the beginning of this chapter shows the front and rear cams assembled in the late 2000-06 except 2006 Dyna models, cam support plate. The individual bearing holes and bearings in the cam support plate supports the front or outboard cam journals (not visible). Inner engine case INA bearings support the rear or inboard cam journals on the end of the camshafts. The top parent-material bearings are just that: the descriptive name uses the alloy support-plate-holes as parent-material bearings. There are no traditional-style bearings. The support plates in the middle and lower picture have actual rolling individual bearings in cups supported by an inner and outer race. There may be a potential problem here to add to a long

list of cam support bearing boondoggles that began in 1999. The outboard cam journal is hardened steel. The parent-material bearing is relatively soft aluminum. The two expand at different rates. The alloy expands faster and more than the hardened steel journals. Specification fitment of journal to parent-material bearing must be done while at ambient temperatures. Under operating temperatures, oil feeds between the two to lubricate their interaction even when the bearing hole expands at a greater rate than the cam journal. On cool-down, isn't it plausible that oil will drain out via gravity force from this enlarged tolerance fitment since a tighter hot-fit is not possible? I would feel better if the Factory cut oil grooves into the interacting face of the parent-material bearing to retain oil for cold start-up. The harder cam journal may damage the parent-material bearing repetitively on cold start-up without initial adequate lubrication. A law of physics called Moment Arm which describes leverage and resultant torsional stress, defines the main problem that surely will cause parent-material bearing holes to oval, particularly the

rear one. The inboard cam-drive-gears on the front and rear cams positioned to one end. If the cam drive gear position in the center of the two cam lobes equidistant from both cam journal ends, no torsional force would occur. However, the drive gear of both cams 1999 to present sits just inside the cam support cover, a fraction of an inch away. The cam lobes and their camshaft extend three or more inches back to their rear journal INA bearing supports. However, with the force applied by the primary and then secondary drive chains, Moment Arm torque will occur transferring its load on the relatively soft parent-material bearing. Why does the energy transfer the short way to the outside parent bearings and not back the long way along the camshaft into the INA rear journal bearings? I surmised that actual INA rear cam-journal bearing rollers would absorb energy by rolling both individually and as a unit. This is why the torsional Moment Arm Force dissipates with the previous late 2000 to 2006 cam support plate front ball bearing and rear cam roller bearing unless the whole bearing can move inside its support hole.

The answer best explains anecdotally. There is a larger force applied to the parent bearing material. Visualize a shaft six feet long and have a person hold each end with the shaft in a horizontal position to the ground. Now, if I grab the middle of the shaft and apply 100 lbs. of pull towards me, their hands/ shaft will move evenly towards me. Now, I will grab the shaft a few inches from the person to the right that represents the (cam support plate), parental material bearing. The other person to the left represents the INA bearing long end. Now I apply 100 lbs. of pull to the shaft. The person to the right (parent-material bearing), will be affected far more than the person to the left (INA bearing), due to the The pictured middle cam-support-plate is the late 2000 to 2006 except 2006 Dyna models that resolved rear cam ball bearing destruction occurring in the original cam support plate. The cure was to replace the rear cam ball bearing with a roller bearing. The 22 offset outboard primary timing chain pulls the rear bearing. A roller bearing can handle this stress but a ball bearing cannot. The front cam up-and-down and sideway

forces are ably served by a ball bearing. The lower cam-support-plate is the original Twin Cam 1999 to early 2000 one. Ball bearings support both front and rear outboard cam journals. The rear cam ball bearing destructed. Cam support plates, cam drive sprockets and timing chains may also be examined in Volume I: The Twin Cam in Chapter I: March 1998: The Introduction of the Twin Chapter II: 1999-2000 Teething Problems and Chapter VIII: The 2007: TC96

Cam Support Plate Bearing Hole Ovaling Is Not New

A thought flashed through my mind.

This same type of occurrence happened before, in the early years, with the TC88s. During many attempts to resolve cam support bearing issues, I believe the Factory experimented with ball and roller bearing fitment in the support plate.

The alternative presumption is that tolerance specifications of bearing fitment in the cam support plate were inconsistent. Many times the bearings would fit into their respective support plate holes with a mechanic's hand press. I say a mechanic's hand press because of the strength accumulated in one's fingers and hands with constant wrenching and all that goes with a mechanic's job.

Since 1999, I have noticed a multitude of cam support plates with scuffing on the inner bearing support holes. I now realize this is the precursor and warning signal that bearing fitment has been too loose and may continue on to oval the hole.

I am always conscious that not everyone is wealthy and can afford replacing every part slightly out of tolerance. The trick is to possess the experience and working knowledge to

know when a part replacement is crucial no matter the financial circumstances and when a minor fix can allow continued safe use of the existing component. In these cases, we at Heavy Duty Cycles will use red Loctite on the outer race into the cam support plate. The mechanic needs to feel confident that the bearing will not move or spin in its cam support plate hole after engine running expansion factors come into play!

Other times, support plate bearing fitment was so sloppy that they would fall out of their holes with a slight thumb push or many times by the pull of gravity when my head wrench Stevie or myself held the plate in a horizontal position. With these situations, we at Heavy Duty Cycles always replace the cam support plate to ensure a bearing press fit.

Now for another interesting progression or should the term be regression, in that loose-bearing fitment and Moment Arm

torsional force caused the bearings, including their outer race to spin in the bored cam support plate holes. This enlarges the holes, leading to a completely new host of problems. I have seen and replaced early cam support plates with egg-shaped support The pull-and-push stresses particularly on the rear cam will cause the borehole to wear asymmetrically, as opposed to increasing the diameter proportionately. The front cam pulls sideways to the back via the rear cam's driving secondary timing chain. The ball bearing is able to cope effectively with this energy. Up-and-down forces apply to both front and rear cams via the weight and movement of the valvetrain. Ball bearings handle this particular stress very well.

The rear cam has some force also pulling it frontward because of the secondary timing chain. However, the aggravating force that destroys ball bearings in the rear cam support plate hole is the 22 offset loading to the forward and

down that is initiated by the driving pinion shaft-timing sprocket. This same force initiates the rear cam-support-plate ovaling with loose bearing fitment. Camshaft Moment Arm torsional force provides an able assist to this process. The combination of loose bearing fitment in the rear cam-support-plate bearing hole, offset loading by the primary timing chain and Moment Arm force is a deadly one.

With proper roller bearing press fitment into the rear cam support plate hole, a roller bearing has withstood the test of time in resisting offset timing chain loading and Moment Arm torsional stress. However, can the relatively soft parent-material bearings in the new style withstand these forces over time? Let us just say that I am nervously monitoring this situation early in 2008.

Moment Arm Force Involves Leverage

In addition to torsional force, a Moment Arm is all about the principle of leverage.

For example, if I put a small weight on the edge of a balanced motorcycle tire and rotate the now unbalanced tire at speed I will create a Moment Arm force. The small weight will extrapolate into a more effective, much larger weight by the centrifugal forces of rotation.

If placing the extra weight in the middle of the wheel, there is no effect on balance, nor does it become a problem through centrifugal force because it is at the center. It simply rotates with the wheel, neither adding nor subtracting energy.

However, the perimeter weight extrapolated by centrifugal force results in a mechanical advantage or, as in the case of the unbalanced wheel, disadvantage. A measured balance

weight placed in the lightest perimeter area of an unbalanced wheel will gain mechanical advantage through centrifugal force to balance the wheel. The mechanical term for this is spin balancing also known as dynamic

## The Principle of Moments

In physics, the Principle of Moments is also known as the moment of force or sometimes just the single word Using the singular word moment may not be a good practice as the Principle of Moments may be confused with a different physical law, the Moment of Inertia. In ancient Greece, Archimedes discovered of the operating principles of levers. The principle actually applies not only to levers but also to pulleys and gears, which is where I am going with this. The Principle of Moments actually describes the process of mechanical advantage or disadvantage, as the case may be.

For our analysis of a cam supported in a parent-material bearing and the effect of the driving force applied to the cam drive gear, this principle of moment will be very telling about potential and possibly premature wear to the parent-material bearing, which is simply a hole in the cam support plate supporting the shaft.

Moment of Force is the force applied to an arm or beam, if you will, or a gear or pulley.

This led Archimedes to develop the formula,

$$M = R \times F$$

Or more simply

$$M = RF$$

M = the moment of force, R = the distance from the applied force to the object and

F = the force applied.

Let me begin to apply this formula and principle to a Twin Cam camshaft. First, if the cam-drive-gear positioned in the center of the two cam lobes equidistant from both cam journal ends, no torsional force would occur and everything would be hunky-dory, at least from a torque point of view.

However, the drive gear of both cams 1999 to present sits just inside the cam support cover, a fraction of an inch away. The cam lobes and their camshaft extend 3” or more back to the rear journal INA bearing support in the rear of the pinion right-side engine casing. In the case of a lever, this distance

will result in advantage (leverage), if lifting a heavy object, making it effectively lighter. However, with the force applied by the primary, and then secondary, drive chains, torque will occur. This twisting torque will transfer as load on the parent-material bearing.

The concept was easy for me to understand but my thinking was backwards or Jeff's was. I needed to understand one more thing: why does the energy transfer the short way to the outside parent bearings and not back the long way along the camshaft into the INA rear journal bearings.

I already surmised that actual INA rear cam journal bearing rollers would absorb energy by rolling both individually and as a unit. This is why the torsional Moment Arm Force dissipates with the previous late 2000 to 2006 cam support plate front ball bearing and rear cam roller bearing

unless the whole bearing can move inside its support hole.

Jeff replied in his easy, anecdotal style, “I have no doubt that there is a larger force applied to the parent-bearing material, and I will give you a practical example. This is a simple example, but bear with me as I think it will be easily demonstrated. “Take a shaft 6’ long and have each person hold one end with the shaft in a horizontal position to the ground. Now, if I grab the middle of the shaft and apply 100 pounds of pull towards me, their hands/shaft will move evenly towards me.”

Then he gets technical for a moment (forgive the pun). “Torque is a straight-line force applied tangentially to a circumference at the center line producing rotation.

Whew!

“Now, I will grab the shaft 6 inches from the person to the right that represents the (cam support plate) parental-material bearing. The person to the left represents the INA bearing long end. Now I apply 100 pounds of pull to the shaft. The person to the right (parent-material bearing) will be affected far more than the person to the left (INA bearing), due to the distance.

Well, his example redirected my thinking rightwards from backwards as Jeff ably demonstrated Moment Arm Force in action with his simple comparisons.

He continued: “I know that the camshaft is short but these forces still apply. The example just helps to make it clearer. I believe it is the unbalanced angular forces (Moment Arm), that exist in their current design, as the cam chain is pictured (2007 Cams in Support Plate) in Chapter VIII in your [first] book.” 6 Jeff was referring to Volume I: The Twin Cam where

the inboard side of a 2007 cam support plate pictured with cams installed shows the timing chain sprockets adjacent to the support plate with the cams' inboard INA-bearing journals extending a long way from the sprockets.

Jeff told me this, just two days after purchasing the book. You have to love mechanics and have extensive knowledge to breeze through the encyclopedic information in The Twin Cam in just a few days. I wrote the book, and I must confess that I cannot do that. I am also gratified that my writings passed Jeff's critical reading.

Metal Expansion Rates and Lubrication

If the metals that comprise the cam journal (hardened steel) that mate with and ride in the parent-material bearing (aluminum) were the same, they would expand at the identical rates as the engine warms up.

The metals are not the same. In fact, aluminum expands faster and more than hardened steel. The cam journal must fit with sufficient clearance when cold. In hot engine running, the aluminum hole will expand more than the corresponding steel cam journal.

When the engine stops running and it cools, there is no more pumped oil supply. Will not gravity cause the oil to drain away from the larger journal to bearing tolerance?

Jeff asked me a rhetorical question, “Regarding the parent-material bearing lubrication, are there oil grooves in the support plate to help facilitate the dispersion of oil on the journal? 7 The answer is no. Wouldn’t this addition to the parent-material bearing surfaces be an improvement that would trap draining oil and give much life to this potentially

beleaguered situation?

All moving parts require a film of oil sometimes only molecules thick to separate them. This separation is what lubrication is really about. Once between moving parts, the film strength of oil is able to withstand great pressures in parts separation and thus lubrication.

On dry start-up, the torsional, twisting force of the levered moment of force will wear the parent-material bearing until oil feed relieves this ongoing problem.

I digress slightly as this is a further compelling reason for using the superior lubricity of synthetic oil. I highly recommend the use of synthetic oil after the break-in period of 1,500-miles in 2006 Dynas and all 2007 and later Twin Cams for this reason alone, never mind its many other

benefits. In fact, I recommend its use in all Harley-Davidson's. Synthetic oil will adhere to surfaces longer and more assiduously than its fossil counterpart can ever hope to. Synthetic oil will prolong the life expectancy of these components. There are many additional and undeniable reasons to use synthetic oil in all modern Harley-Davidson's, which I will explore at length in Volume VIII:

Furthermore, the Moment Arm torsion will not be equaled on all 360-degrees of the perimeter of the parent-material bearing. The torque will want to oval the hole of the parent-material bearing. When parent-material bearings wear out, the enlarged holes will be oval.

Now, the question becomes how fast this process will occur. Jeff thinks that this wear process will happen sooner rather than later. Jeff left me with one last comment, "How many riders would be willing to remove the bearings from their front and rear wheels in place of an aluminum bushing. I

think I will stick with the bearings.” 8

The timing chest, even with all the Factory improvements and upgrades along the way, has been the chief boondoggle to the Twin Cam since its inception and the struggle may not be over.

## Knight Prowler (Silent) Advanced Belt Drive System

### It’s All about Torque

The nomenclature Wood Knight Prowler Advanced Belt Drive System also refers to this belt-gear drive.

This heading may be a little misleading to the uninitiated. When we hear the words belt drive system we automatically

think of primary or rear drive belts that deliver powertrain torque to the rear tire where the rubber matches the asphalt. However, in this case a belt system connects two pulleys that replace the pinion-cam-drive-gear and the rear-cam-drive-gear on the outboard side of the cam support plate inside the timing chest.

Knight Prowler products take their name from the elusive but magnificent and fierce black panthers of the American South.

Bob Wood of Wood Performance Carburetors is all about power. More specifically Bob concentrates on the power that derives from torque. Bob began his H-D aftermarket career by specializing in carburetors with his popular King Carburetors. He naturally branched out into hi-flow air filters and later into cams.

Bob's cam designs are renowned for lowering the powerband into very useful rpm ranges. His grinds use shorter duration profiles, which determine the time the valves are open than most other manufacturers. He increases the lift, which is how much the valves open relative to the duration.

Many of today's aftermarket cams, including Screamin' Eagle, generally have longer duration profiles relative to lift that take away from bottom-end grunt torque for off-the-line power. These longer duration cams work wonderfully in higher rpm stratosphere riding that can challenge the mechanical limitations of pushrod-operated engines.

Do not get me wrong. Duration is a good thing. It is a strong component in the equation for more power derived

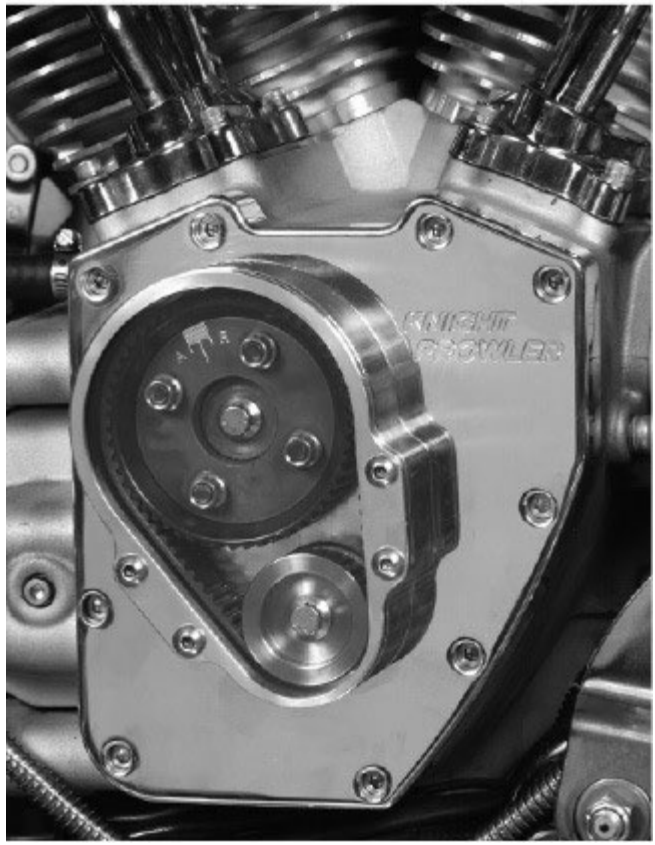
from camshaft lobe design. However, more is not always better. Too little duration as exemplified by EPA-dictated stock cams is not conducive to maximum power either. These cam profiles are all about containing emissions. Power is a secondary consideration.

How do we maximize torque? It is simple. We need maximum cylinder pressure for maximum torque. We get the greatest torque by closing the intake valves early and, secondly, by not opening the exhaust valves too early. We do not want potential compression leaking off because valves are open at the wrong time. Maximum cylinder pressure occurs approximately between and after top dead center (ATDC) of the power stroke.

Although the piston is descending on the power stroke or as Bob Wood says “running away from TDC”, the burning gas is still creating lots of expanding heat to keep driving the piston down.

So, make no mistake; Bob Wood is totally into power. He is a two-time NHRA Fuel Funny Car national record holder.

Knight Prowler (Silent) Advanced Belt Drive System



Pictured is an installed Knight Prowler belt-gear drive assembly on a TC88 engine that replaces the timing chain apparatus in the Twin Cam timing compartment. Bob Wood is the inventive and knowledgeable man behind the Knight Prowler Advanced (Silent) Belt Drive System, which is a progressive step beyond the rest of the aftermarket in this area of timing chain replacement. Bob's really trick and beautiful kit is comprised of outboard primary-belt-driven pulleys with inboard secondary gears connecting the front and rear cams. Bobby's inboard gear drive cams come with either spur gears or helical-cut gears. Helical gears are the quietest available since they are in contact all the time, thus eliminating backlash noise. On a stock engine set-up, two sprockets connect by a timing chain. The smaller sprocket attaches onto the end of the pinion shaft, which extends through the cam support plate. The gear hold-down bolt screws into the end of the pinion shaft. The larger sprocket bolts into the end of the rear cam journal sitting in its cam support plate bearing. The longer primary timing chain

tensions by a problematic spring-loaded adjuster from 1999-2006, or by an improved, actuating hydraulically nylon tensioner pad beginning with the 2006 Dyna. Bob Wood's carbon fiber belt and pulleys replace this primary timing chain system. The inboard cams and their drive sprockets are behind the support plate. The sprocket on the rear cam connects with an identical sprocket on the front cam via the shorter secondary timing chain and tensioned by the same methodology as the primary outboard chain. The secondary timing chain apparatus replaces by either a spur gear drive or an optional and quieter helical gear drive. A special feature is that the rider or mechanic can advance or retard the camshafts a full in about ten minutes. This will adjust the power and torque curves up or down to suit individual riding styles. The Knight Prowler incorporates a unique one-piece 5mm steel billet crank hub with a removable flange for quick installation or service of the timing belt or any other parts related to disassembly. Other highlights include a special hard anodized aluminum 5mm cam gear and steel billet adjustable

spider gear; one-piece billet aluminum face plate with Viton seals and a quick removable belt guard with a clear Lexan cover that you can view the belt drive in action. Goodyear designs the super hi-torque carbon fiber reinforced 5mm belt exclusively for Bob Wood. The Advanced Belt Drive System is available in polished or chromed. I have a chromed one and it has an exquisite lustrous finish. Minor bent, out of true pinion shafts or scissored flywheels; whatever way chosen to describe this all-too-often Twin Cam malady often generates excessive valve noise and can make parts assembly difficult. The belt drive will alleviate this type of noise and make installation easier.

Advanced Belt Drive System

The Knight Prowler Silent Advanced Belt Drive System is an innovative gear drive upgrade for the Twin Cam timing chain arrangement.

The Knight Prowler cures the demonstrably fragile 1999-2006 timing chain spring-loaded adjusters. It will also replace the good looking but still unproven, newly designed chains and hydraulic adjusters in the 2006 Dyna and all 2007 and later models.

However, this is where the similarity to other gear-driven cams manufactured by the majors -- Andrews, Crane, and S&S -- ends. The unassuming but very talented inventor Bob Wood took his gear drive system much further than his predecessors did.

When I initially complimented Bob on his superb designs, he shrugged my comments off, saying, in effect, “There is nothing new here; we’re just applying simple proven automotive technology. It’s old, old stuff; other companies

have been doing it for thirty years.” 9 Well, it might be old stuff all right, and it is true that cars have used these technologies for a long time, but what Bob might not get is that no one else thought to use these concepts on a Harley-Davidson. Nor did anyone design these ideas in a beautiful, workable, functional, and reliable way to up the performance capabilities of our Harley’s.

Adjusting the Powerband

The cam timing is adjustable on the Advanced Belt Drive System. The camshafts can have cam timing advanced or retarded a full in about ten minutes. This will adjust the power and torque curves up or down to suit individual riding styles.

Powering Bob Wood’s gear drive cams on the inboard side of the cam support plate are either spur-type gears or helical

ones. I will discuss these gear types much more as we proceed.

## Spur Gears

Spur or straight-cut gear drives are nothing new. Their use to drive Harley camshafts goes back forever. Every Harley used them before the advent of the Twin Cam in 1999, which replaced gear drives with timing chains and sprockets. Spur gears are not in constant contact, which leads to a condition called backlash defined by John Andrews of Andrews Products as “Whenever the roller follower on a cam lobe passes the maximum lift point, the forces on the (spur) cam drive gear tooth change direction. If more than .002” backlash is present between the cam and spur pinion gear, this directional change of force will result in an audible click as the backlash moves from the backside of each gear tooth to

the front side.” 10

Modern spur gears used by the likes of Andrews, S&S, Wood, and Harley-Davidson are extremely precise. Modern spur gears make the least noise of any over time. There really are no further precision gains available under present manufacturing technologies.

What is new about Bobby Wood’s design is using inboard cam support plate helical gears as an option on the camshaft lobes themselves.

Helical Gears

Helical cut gears are much quieter than spur-type gears because there is continuous contact between mating gears. Helical gears eliminate the clicking, clattering sounds

resulting from spur-gear backlash. As Bobby Wood says, “Helical gears have been around for some time and are not new to the automotive industry. Automotive transmissions have been using helical gears for decades for a smoother and quieter transmission. The motorcycle industry is just now beginning to realize the vast improvements gained by using helical transmission gear technology. Helically designed gears function with less noise than traditional gear designs as they roll smoothly together, not producing the slapping sounds that are common with straight-cut gears. Another benefit in using helical gears is that they have more contact area and oversize or undersize gears are not necessary. Being more complex and difficult to manufacture, helical gears are more expensive. Thousands of phone calls over the years regarding whining or noisy gears prompted Wood Performance to reevaluate the entire noise issue with Twin Cam engines, not to mention the huge problem with slightly bent or out-of-round twin-cam crankshafts. Complicated? Maybe, Wood Performance has a unique simple solution: belt drives on the external gears and

helical gears on the inside.” 11

Bob goes on to say that; “Knight Prowler helical gears are made only with aircraft quality 8620 steel. Besides being ground to extremely close tolerances (.0001”), the helical teeth are ground with a crown, meaning a convex crown of approximately .0002” is ground into each tooth on the rear gear, which results in a much smoother and quieter engagement of the teeth. Silence is golden! No expense was spared in the manufacturing of these one-of- a-kind crowned helical gears. We are always looking for a better way! Any input or ideas to make them better is always welcome here!” 12

Hmm? Bob said, “Nothing new here. We’re just applying simple proven automotive technology.” Well, yes, he does but sounds to me like he has some very skilled inventor traits in

him also.

Helical Cam Gears

Pictured are the front (F) cam helical gear and mating rear (R) cam helical gears. Helical gears have been a fixture in the automotive industry for decades. They reduce noise by eliminating clattering backlash, an ongoing problem with traditional use of straight-cut spur gears in Harley-Davidson

transmissions and timing compartments. Smoother operation results from helical gears rolling in together as they mesh. It is only recently with the introduction of the 2006 Dyna six-speed transmission that the Big Twin begins to take advantage of these and other benefits. The oversize-undersize spur gear fitment procedures common for noise reduction are unnecessary with helical mating gears because of their inherent fuller and constant contact with each other. Wood Performance presents a viable, reliable, and functional alternative to H-D timing chains from an EPA-noise-reduction perspective. Furthermore, the use of the helical inboard cam-support-plate cam-drive-gears in combination with a belt drive replacing the outboard cam support plate rear cam drive gear and the pinion drive gear and timing chain offers performance benefits impossible with timing chains. Hence, the exciting development of the Knight Prowler Advanced Silent Belt Drive System combines these two drive features. As Bob Wood says, “Complicated? Maybe, but Wood Performance has a unique simple solution: belt drives on the

external gears and helical gears on the inside. Nothing new here – we're just applying simple proven automotive technology to our beloved Harley-Davidson motorcycles." 13

In my opinion, the combination of quieter inboard helical drive gears and outboard cam support plate belt-driven pulleys powered off the pinion shaft-end can achieve EPA noise restriction guidelines.

The bottom line is that the Advanced Belt Drive System is precise.

Finally, it is beautiful and trick in every sense of the word.

Whether using inboard cam support plate, spur, or helical gears, Bobby's belt-driven cams achieve both form and function.

The Advanced Belt Drive setup is a definition of Harley cool.

## Belt Reliability

Readers may be asking themselves if the actual belt driving the Knight Prowler cams is a fad design and reliable over time? I confess, I wondered much the same. If I did not respect Bobby Wood's intellect, knowledge base, and design capabilities, I might not have even tried his Advanced Belt Drive System.

Do not get me wrong; I loved the look. The beautiful Knight Prowler chrome timing cover with a protective clear plate over the aperture in which to view the cam support plate outboard drive pulleys and belt in action or at rest is stunning. There is an optional chrome cover available to conceal this aperture, which is also attractive. To me, I would never cover the action underneath.

Let us get back to the initial, as it turns out, ill-conceived, belt reliability concern. Like, who wants to break a belt in the middle of nowhere on a cold, rainy night? Does it get any worse? Even if you were in the big city, where would one find such a belt in stock at the local bike shop? Well, I trusted Bob Wood and certainly did not want to risk insulting him by asking him for a spare belt to carry around.

I waited to ride my hopped-up 103” high-compression 2007 FLHXI Streetglide outfitted with a Knight Prowler Advanced Belt Drive System with big .650 lift high-torque Wood cams for about 4,000 miles before writing this section.

I do not always follow my own advice, sometimes to the detriment of my wallet. Pretty soon, I was redlining the engine in first gear off the line and winding up the revs with highway passing, just having a ball. I break in my bikes and warm up the engine and drivetrain before becoming

aggressive, and I change the oil probably more than need be. However, after these very important considerations I make my bikes honk.

So, how did the belt hold up? It still looks new with nary a wear mark. The belt did not stretch. Zero problems are my experience. In fact, I stopped having a concern about it a long time ago. There is one caveat as always. There must be proper belt installation and sprockets correctly aligned but then again, so do timing chains and their sprockets as do other manufacturers' gear drive systems.

What makes this belt so special? Carbon fiber and precise manufacturing of component parts are the short answers. As stated on the Bobby Wood web site (www.woodcarbs.com), the belt is a "New proprietary 5mm Super High Torque CARBON FIBER REINFORCED belt, designed by Goodyear

exclusively for our Knight Prowler Advanced Belt Drive.” 14

## Cam Timing Variance in Review

Although the Knight Prowler Advanced Belt System is half belt drive located on the outboard side of the cam support plate and half gear drive located on the inboard side of the cam support plate all the gear drive benefits, and then some, apply to the Knight Prowler.

S&S Cycle Inc. first replaced stock Twin Cam timing chains with a gear drive system shortly after the introduction of the Twin Cam. Gear-driven cams in the Twin Cam engine first allowed the utilization of higher seat-pressure valve springs without exacerbating cam-timing wandering. This, in turn permitted use of much higher lift cams without loss of some power due to imprecise cam timing.

The loose fit of the two timing chains, primary and secondary can result in up to of cam timing inconsistency. The primary chain connects one sprocket on the outboard side of the cam support plate to power the rear cam, from the pinion-shaft-drive-sprocket. The secondary chain is on the inboard side and drives the front cam sprocket via the rear cam sprocket. The cam timing can continuously shift back and forth through this variance. Furthermore, in performance applications, these higher seat-pressure valve springs can affect cam-timing wander an additional to simply because the strength of the valve springs is greater than with the 1999 to 2006 spring-loaded cam chain tensioners.

This effect is less on the new hydraulic cam chain adjusters introduced on the 2006 Dynas and carried over into all Twin Cam models in 2007 to present. However, this effect is constantly at work and will exacerbate as the hydraulic action of these late-style tensioners wear over time.

Indisputably, cam timing cannot wander at all with a properly fitted, all-gear drive arrangement or the hybrid gear-belt Advanced Belt Drive System.

A second major benefit of gear drive cams is that much higher lifts are possible without mating cam lobe interference with the cam lobe on the opposing cam.

Development of this feature for the high-lift timing chain cams also followed with an effective but different methodology…changing the cam base circle dimensions.

Powering Gear-Driven Camshafts

A gear drive set-up has two mating gears on the outboard

side of the cam support plate. The first runs off the end of the rotating pinion flywheel shaft. This gear bolted and slotted to the pinion-shaft end powers -- or more correctly -- transfers power to drive the whole valvetrain. This drive gear meshes with a larger gear bolted and splined to the outboard end of the rear cam journal that sticks through the cam support plate. Thus, the rear cam now rotates via these two drive gears. The rear cam inboard gear meshes with the front cam inboard gear to rotate it also.

Timed Engine Components Coordination

These gears time with piston position in the cylinders via the pinion shaft. The cast pinion shaft is part of the right-side pinion flywheel, which rigidly fixes as one, well, least it did until 2007 problems with scissoring flywheels…with the left-side flywheel via the .007” pressed-in crankpin.

The connecting rod big ends rotate on bearings and cages on the crankpin. At the other end, the connecting rod small ends house the wristpin bushings, which attach the pistons to the rods with wristpins.

Many also refer to the flywheels, shafts, crankpin, bearings, and con rods as the crank or flywheel assembly.

The cam lobes, two on each cam journal, operate the four valves, two in each head. The elliptical lobes open the valves, timed relative to piston location. The valve springs close the valves as the rotating cam lobes diminish in height.

A magnetic sensor called the crank position sensor in the front left-side engine casing reads notches cut into the left-side flywheel, also known as the sprocket side or power side flywheel. The sensor sends magnetic pulses to the electronic

control module (ECU or ECM) at the heart of the engine management system. It may be easier to visualize the ECM as the brain. The pulses tell the brain where the flywheel rotation is relative to piston location. Flywheel speed, among other factors, will indicate when the brain will fire the spark plugs to ignite the fuel mixture.

Gas mixed with air needs time to start the burning process. Combustion lag is the name for this.

The pistons move in tandem with the flywheels. If pistons are moving up and down the cylinders faster, this indicates high rpm. The ECM brain will consider this and spark (fire) the spark plug sooner to enable earlier combustion. In other words, the spark will begin fuel ignition while the piston is still on its way up the cylinder on the compression stroke. The goal is to have a full flame front traveling across the combustion chamber as the piston reaches its top dead center (TDC) on its compression stroke to drive the piston back down

the cylinder on its power stroke.

The burning fuel creates heat. Heat expands to create pressure. Pressure drives the piston down on the compression stroke.

Timing advance ignites the fuel sooner before the piston reaches TDC on its compression stroke. As rpm increases, advanced ignition timing also increases. Advancing timing sparks the fuel mixture progressively sooner during the compression stroke. As rpm decreases, ignition retards because the fuel will have more time to burn before the slower-moving piston reaches TDC. Retarding ignition timing simply means the ECM sparks the ignition process later with the piston progressively closer to its top dead center travel. This is what I mean by ignition curves.

The mechanic can advance or retard other events in an engine such as cam timing, cam lobe separation angles and cam lobe centerlines, which I will examine in detail Chapter IV: Cams, Cams, Cams subsections, Valve Overlap, Cam Lobe Centerline Angle (LCA), and Cam Lobe Separation Angle (LSA).

The point is that every moving part in a Harley-Davidson engine times to each other for very specific reasons in order to open intake valves to deliver fuel into the engine, ignite the fuel, drive the pistons down to create torque and power and then get rid of spent gases out the exhaust valves. An engine is all about parts timing to each other for proper sequential operation.

How mechanics design or play with the timing of this process will determine emissions and torque either in opposition or in tandem with each other.

# Which Cylinder Is the Boss?

Some of the following may sound confusing at first, so please be patient, and think it through. Part of the perplexity arises because Harley-Davidson and some in the automotive industry view this subject of which, is the lead cylinder from opposite vantage points. However, it is all really the same thing. The seemingly opposed but same subject view translates into confusion of language.

A four-stroke engine like Harley-Davidson requires of crankshaft rotation to complete all four cycles or strokes. They are intake, compression, power, and exhaust, each lasting crank rotation. One might think that each cylinder operates apart from the other. If the cylinders were opposed, this would be true. However, there are angled cylinders.

Therefore, the two cylinders fire awkwardly, but that is life in the wacky world of Harley, where engineers in 1903 felt a engine would not even run.

Bobby Wood explained to me that one cylinder fires at while the other fires later.

The confusion compounds because Harley-Davidson views the number one cylinder as the front one with the rear being the number two cylinder. The automotive industry, which is obviously huge compared with the Motor Company, views the rear cylinder as the master cylinder and therefore the number-one cylinder. This is the logical way to look at this because the rest of the engine times off the rear cam lobes and the rear cylinder.

Logic is not always the way with our beloved dinosaurs.

Therefore, the automotive sector calls the front cylinder the number two, relegating it to the slave cylinder status because its activities take its timing from the master rear cylinder.

The reader needs to know who is doing the explaining to envision what exactly is happening.

Therefore, from Harley-Davidson's perspective, when the front number one cylinder fires, the rear number two cylinder will fire at crankshaft rotations later. The front number one cylinder will fire again at crankshaft rotations later, and on it goes.

If the reader adds and the crankshaft rotation, the two figures will total the complete crank rotation required to

fulfill all four strokes.

The auto industry along with Bob Wood quite logically views the rear cylinder as the number one master cylinder because it drives the front cylinder cams. Therefore, when the number one rear master cylinder fires, the front number two, slave cylinder will fire later. The number one rear master cylinder fires again afterwards, thus completing the required crank rotations to fully complete the intake, compression, power and exhaust cycles. 15

If the reader is perplexed, the real problem is one of semantics. How confused will one become not knowing the above when an authority like H-D postulates mechanical know-how within the confines of its interpretation in literature and when other experts in the field postulate within the other interpretation?

The quest for less noise compelled by EPA dictates drives new methodology and design. After all, the Twin Cam's dual cams, driven by silent cam chains, design strategy was all about reducing gear-driven timing chest noise.

However, it just keeps coming, as the strictest noxious exhaust emissions controls to date will introduce in 2010.

Water jackets and radiators will be two of the next Factory responses to quelling noise even more as the government mandates further restrictions. I dislike being alarmist especially with H-D traditionalists, but I view the Twin Cam as a transition engine from the past to the future. Water jackets and radiators; catalytic converters, overhead cams eliminating our beloved valvetrains; changing the traditional

45-degree cylinder angles to other more compatible angles like those of the V-Rod or the new 1125cc Buell will all occur until we reach alternative fuels.

Form vs. Function

Form is good looks while function is all about workability. The Knight Prowler Advanced Belt Drive System is the perfect marriage of form and function, no mean feat.

However, there are two other related reasons driving H-D design primarily by the symbiotic aftermarket. The first is form over function whereby V-twins are individualized and personalized. This is all about

Well, the Knight Prowler's visible belt and pulleys through a clear Lexan glass surrounded in a sea of lustrous chrome is a definition of cool. It looks so nice on my bike that even

seasoned riders do not comment but just stare as if they understand what is going on. In my neck-of-the-woods, in and around Toronto, I am the first to have an Advanced Belt Drive System and other riders are in awe wondering what Petersen is up to now.

With other products, many times cool will compromise functionality of the bike. Lowering a Harley so much that shocks and forks cannot work as originally designed is an example. Another is raking the frame and extending the forks to the point where they compromise safe riding. Both examples involve extreme design form over function.

Compromising function may not occur with minor lowering or conservative rake and fork length changes. In fact, it may assist handling and look good at the same time.

A shorter rider will have more control with slightly lowered shocks, forks, and seat. I am not a small guy but before H-D began lowering all of its bikes to accommodate the average-sized rider, there were many models I could not sit on flat-footed.

A police department may extend forks a few inches on pursuit bikes for more high-speed stability. This is an example of form assisting function.

Pure functionality overriding form usually leads to ugly but the bike performs wonderfully.

Harley riders – I am one myself -- are forever seeking a Many also seek performance. Hey, that is I, too!

A look is form, whereas performance is function.

Therefore, the question becomes how to get rid of those bothersome stock Twin Cam timing chains, chain adjusters, and bearing problems. How do we control wandering cam chain timing caused by loose fitting sprockets and timing chains further exacerbated by the pulls and pushes of the valve springs versus the timing chain adjusters?

How about combining form and function to eliminate these aggravating situations, also gain much performance, and create some wild form. When Bob Wood designed the Knight Prowler Advanced Belt Drive System, I just had to put it through its paces in aggressive city and highway riding over last summer. Bob's specialty is high-lift, high-torque cams producing oodles of power right out of the gate, and he is no slouch in the design of good looks department either. I am looking very good, I am going very fast, and, by the way, that is fast and reliable. I will be writing much more about Bobby Wood in Chapter IV: Cams, Cams, Cams since this guy is a wealth of invaluable information.

# The Advantage Summary of Gear Drives over Timing Chains

1. The first advantage is reduced maintenance since there are no chain guides or tensioner shoes to wear or replace.

2. Elimination of debris from wearing parts especially 1999-2006 adjuster pads and potential 2007 and later parent-material bearing ovaling in the oil is a big advantage.

3. Cam timing is constant and accurate. Cam chains can stretch while the gears remain stable. There is a reduction of bearing loads since no spring-loaded cam tensioners are used on 1999-2006 models.

4. Use of higher cam lifts is possible with the gear drive

since the gear cams rotate differently than the cam-chain cams. This allows more space between rotating parts.

5. The rider does not have to worry about ball bearing failure on early Twin Cams. The gear drive alleviates the different stresses induced by the stock cam chains and their tensioners. The rear cam roller bearing, which is necessary to combat the higher-pressure cam chain tensioners, will elevate gear noise. The quieter gear drive ball bearing solves this emissions issue.

Accuracy dictates that I remind the reader that overall, a spur-gear drive should produce more emissions noise than timing chains, but, in practice, this is not always the case.

In the example of Bob Wood's belt drive helical gear Knight Prowler system; he has resolved this noise issue while delivering mucho performance.

Therefore, there are three main reasons for considering the switch away from the timing chain-driven cams to gear-driven cams.

1. The first is for the efficiency minded.

2. The second is for the performance oriented.

3. The third is to correct potential mechanical breakdown from the stock components.

## Gear Drive Installation Instructions 1999-2006

Use these directions in conjunction with your H-D workshop manual and the instructions supplied with the

geardrive cam kits for 1999-2006 models except the 2006 Dynas.

General Installation Advice

Instructions for 2006 Dynas and all 2007 to present Twin Cams are essentially the same but vary from these directions because hydraulic chain adjusters replace the earlier spring-loaded ones; the cam support plate uses parent-material bearings; spacing and torque values may be different and the oil pump is updated.

Therefore, procedures that accomplish the same goal -- to install a gear drive -- are different.

1. Always disconnect the battery when performing engine or powertrain work. Do not wear metal rings. Inadvertently engaging the starting system and/or a shorting metal ring may

cause painful injury and/or parts damage.

2. On Twin Cams, the covers are not load bearing like predecessor engines. Remove the timing cover following the instructions in the shop manual.

3. Using the Kent Moore cam chain tensioner unloader tool (HD-42313), retract the primary cam chain tensioner. Repeat this procedure for the second tensioner after removal of the cam plate. The retention pin part of the tool inserts into the support plate to hold the springs and tensioner pad away from the timing chain.

4. Remove the flange bolts and flat washers from the seventeen-tooth crank sprocket and the thirty-four-tooth primary cam sprocket. Use the crankshaft/camshaft sprocket-locking tool (HD-42314). This locks the two sprockets so they

cannot turn during disassembly.

5. The seventeen-tooth crank sprocket drives the whole valvetrain as it attaches to the end of the rotating pinion shaft. Gently ease off the primary cam sprocket and the crank sprocket. Remove these with the primary cam chain as a unit.

6. If doing normal service and using the chains again, mark the chains' outside plates to ensure they reinstall in the same direction they came off.

7. Release the cam support plate from the oil pump flange by removing the four Allen head socket screws.

8. Then release the cam support plate from the crankcase flange by removing the six Allen socket screws.

9. Now ease the support plate from the end of the

crankshaft. It is not necessary to remove the oil pump from the pinion shaft.

10. Examine the 1999-2006 except the 2006 Dyna cam support plate to ensure a roller bearing supports the rear cam. This is an indicator that the desirable later cam support plate is in place. In a gear drive, a ball bearing replaces the roller bearing. A ball bearing will still support the front cam.

11. This support-plate part number should be 25267-99B. If the bike has an earlier support plate with a different part number and two ball bearings, it may be prudent to upgrade to the latest cam support plate even though gear drives utilize a ball bearing, front, and rear. I always make this upgrade when finding earlier support plates identified by the part numbers 25267-99 and 25267-99A.

12. The later parent-material-bearing support plates do not need replacement unless cam-journal-fitment tolerances are beyond H-D specifications. It is important to check for this because Moment Arm Force may have begun ovaling the support holes.

13. Remove the bearing retaining plate from the back of the 1999-2006 25267-99B-cam support plate. Use the Kent Moore camshaft remover/installer tool HD-43644, to press both camshafts and bearings from the support plate at the same time. After removal of the stock cams from the cam support plate, you may discard the chain tensioners, as they will not be required with the gear drive.

14. Always use new cam bearings in the support plate. If the bearings are a loose fit in the plate, replace with a new 25267-99B-support plate.

15. Remove the INA camshaft needle bearings from the crankcase using bearing/remover tool HD-42325.

16. Also, replace the engine case cam bearings, substituting Torrington B148 bearings for the stock INA SCE148 bearings. On 2006, Dynas and 2007 to present Twin Cams replace the stock SCE168 INA bearings with Torrington B168 bearings. The later bearings have a 1.000” inner diameter, while the earlier ones are .125” smaller with a .875” inner diameter.

Do not reuse bearings. The S&S Camshaft Installation Support Kit (33-5163) for 1999-2006 supplies new ball bearings and rear journal Torrington bearings.

17. Apply assembly lube to the outer races of the cam bearings and the bearing boreholes in the support plate.

Install new bearings using a hydraulic press and Kent Moore camshaft/remover tool HD-43644. Many times the bearings will fit into their respective support plate holes with a hand press. If so, replace the cam support plate to ensure a press fit. This loose fitment will allow the ball bearings to spin in the bored cam support plate holes, which will eventually enlarge them, leading to a completely new host of problems. I have seen cam support plates with egg-shaped support holes. The pull-and-push stresses on the cams will cause the boreholes to wear this way, as opposed to increasing the diameter proportionately. A law of physics called Moment Arm Force will oval the holes particularly the rear one.

18. Ball bearings must be used front and rear with the gear drive because a roller bearing is inappropriate for the new stresses and will make too much noise.

19. Lube the outer bearing journal surfaces of the cams and the inner races of the bearings.

20. Press the front shorter cam into its bearing first. Use a new retaining ring.

21. Then do the same with the longer, rear camshaft ensuring alignment of the timing marks. During installation, to avoid damaging the ball bearings' inner cups, the bearings require support by the inner races.

22. Camshafts over .600" lift (stock is about .473"), such as the S&S 625G (.625" lift) or the 640G (.640" lift), require the crankcase be relieved a little to provide .030" clearance for the cam lobe rotation. Always check for cam lobe clearances by rotating cams when installed in the crankcase needle bearings.

replace Some O-rings are close in sizing to others. Pay particular attention that correctly sized O-rings are used.

24. Apply assembly lube to all surfaces that move against and with another surface.

25. Align camshafts with and into the engine case Torrington needle bearings and gently slide the support plate over their alignment dowels. Do not force!

26. Use blue Loctite and torque to the lower end of Harley-recommended specifications.

27. Torque bolts alternately in correct sequence as demonstrated in S&S instruction sheets.

28. Install oil pump bolts; torque to 95 in-lbs, alternately sequenced.

29. Using red Loctite on the threads and a drop of 20W50 oil under the Grade 8 bolt flanges, install the crankshaft gear, timing mark out, on the crankshaft first. Torque this bolt to 25 ft-lbs.

30. Put drive gear key in rear camshaft. Place cam drive gear on rear camshaft. Rotate until timing marks align. Use the thick washer supplied and torque to 34 ft-lbs.

31. Check the cam drive gear for interference with the cam cover. Ensure there is .030” play between the two by measuring with clearance putty (Plastocene) and relieving the cover as necessary. The area of concern is between the two top right-hand cam support cover bolts. Place the putty inside the cover. Gently install and remove the cover. Measure the compressed putty. Relieve cover, if necessary. Double-check with putty clearance, check again. Install the timing cover.

Discard stock fixed-length pushrods and install adjustable pushrods. The longer rods are for exhaust valves, the shorter ones for the intakes.

1. Rotate engine until front lifters are at their lowest point.

2. Turn adjuster screw on front exhaust pushrod until there is no up-and-down play but will still allow rotation of pushrod with light finger pressure.

3. Turn screw a further twenty flats which equals 3 ⅓ revolutions.

4. Repeat procedure for front intake pushrod.

5. Allow the hydraulic lifters to bleed down for 10-15 minutes, before proceeding. Do not rotate engine until lifters bleed down or something expensive will bend and/or break.

6. Rotate engine and repeat the above steps for the rear cylinder pushrods.

7. The engine will be noisy on start-up for a few miles as the hydraulic lifters fill with oil and become quiet.

At Heavy Duty Cycles, we always install the S&S limiter lifter rings with Andrews, Crane, Screamin' Eagle, and S&S cams into the hydraulics to ensure hydraulic lifter quietness with solid lifter performance. I do not use them with the different torque-driven Wood cam profiles.

It is extremely important to use the proper tools, removal

and installation procedures, torque specifications, and torquing sequences. I cannot overemphasize this. Twin Cam engine work is not for the novice. The shop manual is mandatory reading. If your mechanic does not have all the TC88 or TC96 tools, he cannot do the work competently.

With that out of the way, is the S&S gear installation easy? Sure it is. In fact, it is much easier than installing the stock components.

Is it quiet? In theory, it is not supposed to be quieter than the stock timing chains. In practice, my experience is that gear drives are sometimes noisier, sometimes the same, and sometimes quieter than the stock timing chain apparatus.

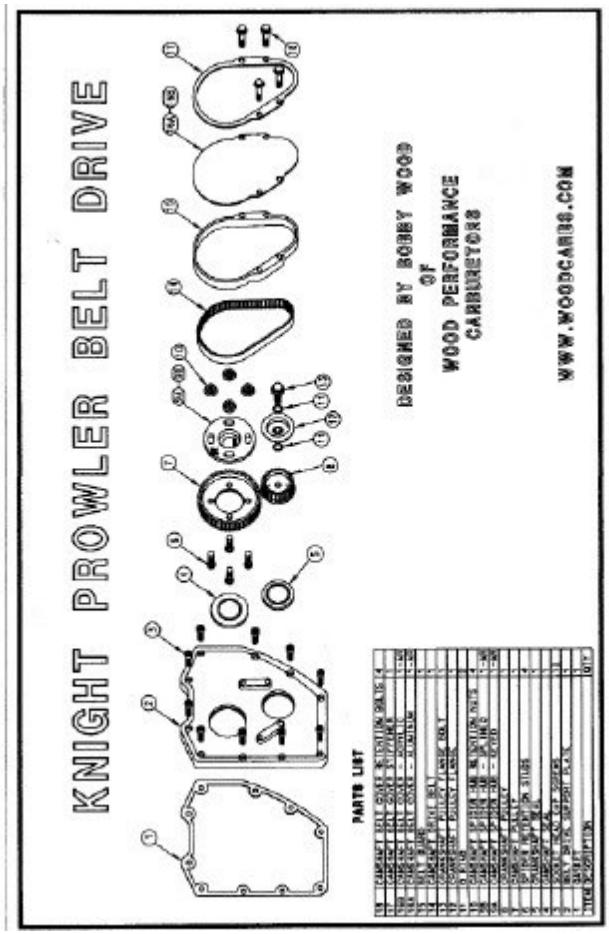
Disassembled Knight Prowler Advanced Belt Drive System

Top right is the gorgeous Knight Prowler (Silent) Advanced Belt Drive System timing cover that replaces the stock TC88, TC96 or CVO 110. It is not a replacement for the cam support plate. The pulleys and carbon-fiber belt fit into the front

aperture. Covering this aperture is a clear Lexan cover (not shown) through which the rider can view the turning pulleys and belt. There is an optional chrome cover. Between the timing cover picture and the camshafts are cam shims for spacing. Andrews Products makes the two cams to Bob Wood specifications. The top cam is the front cam while the lower one with the keyway is the rear cam. To their left is pair of inboard cam-support-plate spur cam gears. Optional helical cam gears are available. I show the helical cam gears in the previous picture in this chapter. The combination of the outboard cam support plate belt and pulley drive, along with inboard helical gears, will make this complete unit a hit with the customizing and performance crowds as well as the EPA. Below the cams are the inboard Torrington B148 (1999-2006) or the B168 (2006 Dyna and 2007 to present) cam bearings depending on year engine that replace the stock INA SCE148 (1999-2006), and INA SCE168 (2006 Dyna and 2007 to present), cam bearings. Below, lower left, is the unique one-piece 5mm steel billet crank hub with a removable flange for

quick installation or service of the timing belt or any other parts related to disassembly. Other highlights include a special hard anodized aluminum 5mm cam gear and steel billet adjustable spider gear. Goodyear designs the super-high-torque, carbon fiber reinforced 5mm belt exclusively for Bob Wood. The cam timing is adjustable on the Advanced Belt Drive System. The camshafts can have cam timing advanced or retarded a full in about ten minutes. The timing marks are visible on the larger rear-cam-drive-pulley opposite the keyway. This enables the mechanic to adjust the power and torque curves up or down to suit individual riding styles. Bottom right is the tools Bob Wood supplies with every kit for ease of installation and removal.

Advanced Belt Drive Diagram




Pictured in diagram form is the sequential parts order of the

Bob Wood timing cover, outboard cam support plate pinion drive pulley, rear cam drive pulley and 5mm carbon-fiber belt, hardware, and gaskets. This unique design not only works wonderfully but also is a great customizing attribute for any engine.

# Knight Prowler Belt Drive Installation, 1999 to Present

Please refer to your appropriate year and model Harley-Davidson manual for disassembly instructions.

If using the Advanced Belt Drive outboard support plate pulleys and Knight Prowler timing cover with stock cams and inner timing chain, disassemble to the support plate leaving, it intact with the engine.

If replacing the stock cams with a gear drive setup, whether driven by S&S spur gears or the Wood helical gears, also disassemble the cam support plate and oil pump, removing the two inner INA cam journal support bearings from the inner pinion crankcase using the proper Kent Moore tool (H-D43644).

It is necessary, in my opinion, to measure pinion shaft end runout with a dial indicator micrometer. H-D has been having problems with flywheels scissoring. This occurs when one flywheel shifts in front of the other. The Motor Company writes that a rider on a rubber-mount H-D cannot feel up to .020” runout on the sprocket shaft in Service Bulletin TT324. 16 This thought makes me vibrate and shudder. Many mechanics including Bob Wood and myself are ecstatic with 0.000” runout on the end of the pinion shaft, happy with .001”, but dislike anything above this although we tolerate .002”. The consequence is removing the flywheels to replace or re-true. The expense of this is onerous, so we reluctantly now have to tolerate up to .004” runout of the pinion shaft.

The good news is the nature of a drive belt system is it will tolerate higher runout figures over inflexible outer drive

gears. However, this advantage does not help runout-induced vibration, or the pinion shaft reaming out the shaft bushing in the cam support plate that the pinion shaft passes through. This reaming creates a looser tolerance between the shaft and bushing that affects bottom-end bearing assembly oil-feed pressure. Oil is pressure-fed from the oil pump into the cam support plate, over to the pinion bushing and fed into the pinion shaft oil passage for delivery to the hollow crankpin and then fed out three holes into and around the three sets of bottom-end rollers.

As I have said many times, in any performance work or simple repairs inside the timing chest, it has become standard aftermarket industry practice to replace these INA inboard bearings with Torrington ones. The part number for these inboard .875” inner-diameter cam journal bearings for 1999-2006, except the 2006 Dyna series, is H-D 9198 (INA SCE148) and should replace with Torrington B148 bearings.

The part number for the same style but with larger inner-diameter one-inch bearings for 2006 Dyna and all 2007 to present Twin Cams is H-D 9215 (INA SCE168) and should replace with Torrington B168 bearings. S&S (33-5163) bearing replacement kits for 1999-2006, except the 2006 Dyna series, contain the timing chest gasket, two Torrington inboard journal cam bearings, two outboard cam journal ball bearings and a heavy duty bearing retainer (33-5212), plate to replace the not-so-strong H-D retainer for the cam support plate. S&S (33-5240) bearing replacement kits for the 2006 Dyna series and all 2007 to present Twin Cams contain the timing chest gasket and two Torrington inboard journal cam bearings.

There are no replacement outboard cam journal ball bearings since they are parent-material. Therefore, there is no need for a heavy-duty bearing retainer plate since there are no actual bearings per sè in the cam support plate.

There are two flat rectangular hydraulic block off metal plates used after eliminating the hydraulic chain adjusters.

In addition, a flat washer is included for the gear drive assembly.

For more information on this subject see The Twin Cam, Chapter III: Timing Chest Breakdowns and Solutions and Chapter VIII: The 2007 TC96

1. Install any camshaft with cam lobe lift over .510” into the inboard cam journal Torrington bearing and rotate to ensure there is a minimum of .030” rotating clearance with any obstructions in the timing engine case. Each cam can be rotated individually by hand a full Late 2006 and later Twin Cam casings have more clearance area cast-into them at the foundry than 1999-early 2006 Twin Cam engine casings.

2. Ball bearings for both front and rear outboard cam journals in 1990-2006 Twin Cams except the 2006 Dynas, which incorporate the 2007 to present changes, are the appropriate choice for use with a gear drive. The stresses imposed on a gear drive are somewhat different and less complex. The timing chains, particularly the angled outboard primary chain, feel additional stress from a gear drive. This extra stress requires a roller bearing instead of a ball bearing for the rear cam outboard support.

The 2006 Dyna-2007 to present models has parent-material support outboard bearings that are not replaceable...like, how do you replace a hole?

It is very important to follow the Knight Prowler Belt Drive installation instructions for a comprehensive setting up procedure. This is all new stuff that even experienced H-D mechanics will find different. It is not difficult, but first we

must walk before we run.

I am briefly going to describe selective and salient aspects of the install, which will be unfamiliar to the reader as the advanced belt system is very progressive relative to what else is available. Do not assume these are comprehensive directions. The H-D service manual and Wood Performance instructions are both necessary for full information.

- The two mounting bosses that secure the black front chain guide (39965-99), not to be confused with the timing chain adjuster must be machined or ground down on 1999-2006 outer cam support plates to the even surface of the cam support plate.

Now proceed with gear drive installation.

1. Install the inner front and rear S&S gears, if using spur type, on both front and rear camshafts. The gears assemble by pressing camshafts into the drive gears with the drive keys in place.

2. The timing marks on both front and rear gears must correctly align. It is an easy procedure and I only mention it because if using the helical gears the procedure is a little tricky before doing it the first time.

3. The angled cut of helical gears prevents a straight press-on action as with straight- cut spur gears. The mechanic needs to anticipate the amount of movement of the interacting gears to press together in correct timing mark alignment.

4. Therefore, initially press the front cam in the support plate only about ¼".

5. Now install the rear camshaft, anticipating movement of the other mating front helical gear. Spur gears mate without movement whereas helical gears will turn each other as they interact. The goal, of course, is to ensure the timing marks line up when both cams fully install in the cam support plate.

6. Wood Performance lists a procedure for checking for sufficient gear lash between the inboard cam spur gears. Lash of 0.000” is too tight; it will cause a whining noise while lash of .002” is too loose, and will cause a clattering noise. The correct lash for these gears is .0005” to .001”. There are both undersize and oversize cam-drive-gears for the spur gears available from Wood Performance or S&S Cycle.

7. On 2006 Dyna and 2007 and later Twin Cams there is a furnished .075” thrust washer to install on the rear cam, after lubricating both sides with a fine grease or 20W50 weight oil.

8. Consult the belt drive diagram. Install the Knight Prowler belt drive base plate cover with the supplied ten new (¼-20 x 1”) stainless steel bolts and new gasket.

9. Torque these bolts to the H-D specification of 90-120 in-lbs in the exact tightening and sequence steps.

10. Install the Knight Prowler belt drive crank-gear-pulley with the outer flange as a unit. Using a ⅜” twelve-point socket, tighten the x 1 ¼”) bolt to ensure the crank gear pulley is seated properly.

11. Remove the crank gear pulley retaining bolt and belt flange. Install the supplied key in the rear camshaft.

12. Install the Knight Prowler belt drive cam gear pulley using the ⅜-24 x 1 ¾” stainless steel bolt with a ⅜” twelve-point head and the 1” heavy-duty washer. Always use this bolt and washer on the initial install, as it will draw the cam gear pulley into the cam while keeping it in alignment.

13. After starting several threads of this bolt, rotate the cam-pulley-drive-gear to ensure even rotation.

14. Each Wood Performance Knight Prowler belt drive kit provides special tools for removing and installing the cam gear and crank gear assemblies.

15. Use a small steel straightedge. Place it on the outward face of the crank gear pulley and check alignment with the outward face of the cam gear pulley. A variance within + or -.010” is within specifications. Supplied shims are available in .005”, .010”, and .020” for use if necessary. The appropriate

shim goes behind the cam gear pulley if it does not protrude enough.

16. Reinstall and recheck for correct alignment.

17. Year 2006 Dynas and all 2007 to present Twin Cams have special thrust washers in gradients between .110” and .150” thick. If correct alignment is not possible with the thrust washer supplied by H-D, the correct one must be purchased to obtain the correct variance within + or - .010”. If requiring further alignment, there are specific instructions in the Wood Performance installation sheets.

18. With both gear pulleys installed but not torqued down, align the cam, and crank timing marks together. This will place the rear cylinder on its compression stroke. Place the timing belt on the lower portion of the crank gear pulley’s

lower teeth or nubs. Hold the belt taut; engage the belt on the right side of the cam gear pulley. Holding the engaged belt teeth, walk the belt teeth around in a counterclockwise direction. This stuff requires a knack. However, it will come easily with patience.

19. Check timing mark alignment once more.

20. Ensure the belt is flush with the end of the cam and crank gear pulleys. Recheck timing marks.

21. Put the supplied O-rings into the removable flange grooves, one on either side. Place the flange into the front of the crank gear pulley. Use a dab of 242 blue Loctite on the ¼", twelve-point bolt and torque to 25 ft-lbs.

22. Now, install the 12-point, ⅜-24 x 1 ¾" cam-retaining bolt with the hardened washer with a dab of 242 blue Loctite

and torque to 35 ft-lbs.

23. Always start with the crank gear pulley advance/retard marks set at stock or the “0” point.

24. Now torque the four 12-point retention nuts to 120 in-lbs.

Thinking mechanics instinctively switch mindset from ft-lbs to in-lbs torque by subconsciously viewing bolt or nut size, thread and metallurgy. Novices have to pay attention because torquing to ft-lbs, specifications when it should be in-lbs. will break or strip nuts, bolts, and mating pieces.

25. Install pushrods.

# Hydraulic Lifter Limiter Kits

I have been a big proponent of limiter kits in the hydraulic lifters to give solid lifter performance with hydraulic lifter quietness when using big-lift cams in both Evos and Twin Cams. The limiter kit contains four steel rings that install inside each of the four stock hydraulic lifters. They limit the travel of the hydraulic tappet plunger in 1984-1999 Evolutions, 1986 to present Sportsters and, of course, 1999 to present Twin Cams.

The S&S Cycle, HL2T spacer rings are the most popular in use today. Part number 33-5338 fits stock 1984-1985 Evos, stock 2003 to present Sportster, S&S tappets for 1984 to present Big Twins/Twin Cams, S&S tappets for 2000 to present Sportsters, and stock 2003 to present Twin Cams. Part number 33-5339 fits stock 1986 to 2002 Big Twins/Twin Cams and stock 1986 to 2002 Sportsters.

I quote S&S's catalog for High Performance Parts for American V-twins Since 1958: "When operated at high rpm, tappets have a difficult time following the cam lobes and therefore may not deliver the intended cam timing. Hydraulic tappets have a tendency to collapse partially making this situation much worse. HL2T spacers do not allow the tappets to collapse so high rpm performance is comparable to that of solid tappets. Another benefit of limiting tappet plunger movement is easier starting. When an engine is not running, the hydraulic tappets tend to bleed down. The effect of this is higher cranking compression when restarting the engine. Some high-compression, large-displacement engines may become very difficult, if not impossible, to start. With the HL2T spacer installed, the collapsing effect minimizes and both high-rpm performance and starting are improved. Under typical driving conditions, normal hydraulic function and the benefits of a hydraulic tappet, minimal noise and maintenance

are retained.” 17

I am still a big fan of limiter kits when using Andrews, Crane, or S&S, cams to name a few. However, they are all long-duration cams relative to their high lifts. The biggest cams I have installed other than Bobby Wood’s camshafts is the S&S 640G gear drive with .640” lift with intake and exhaust duration. These work fine with the limiter kit and perform better.

However, with the .650 lift and 248-degree duration on both intake and exhaust Knight Prowler Advanced Belt Drive System, I tried to use an S&S limiter kit. The result was lots of noise and no amount of tinkering would help. I turned to Bobby Wood for help.

Bobby informed me that the ramp rate design of the cam

lobes and the amount of cylinder expansion growth due to operating heat expansion had more influence on noise levels than lift-to-duration ratios. Different manufacturer cylinders will grow at different rates, some more than others. This could be the situation with my particular build.

Bob Wood told me to take the limiters out and adjust the stock hydraulics down much further than other manufacturers advise. Under duress the limiters will create a solid with this procedure, which is what limiters do.

## Adjusting Performance Pushrods

The total travel in an H-D hydraulic lifter, which, by the way, originates with the small-block Chevy, is .190” to .200”. Adjust the pushrods down .140” to .150”. This will take the unit close to a solid lifter. Extreme riding will bring the pushrod down in contact with the bottom of the plunger

travel, in effect creating a solid lifter of metal against metal with no oil-hydraulic buffer.

How do we get a .140” to .150” adjustment?

Most adjustable pushrods adjust in the range of twenty-eight to thirty-two threads per inch per full turn. JIMS Pro-Lite (www.jimsusa.com), and Crane Time Savers (www.cranecams.com) use twenty-four threads per inch. Rivera Taper Lite (www.riveraengineering.com) uses forty threads per inch. Andrews, Crane, H-D, and S&S all have adjustable pushrods at thirty-two threads per inch. However, do not take this stuff for granted and always check the installation instructions to see how many threads per inch a particular pushrod set is. Manufacturers change specifications and may introduce kits with different treads per inch. As an example, if thread-per-inch specification is thirty threads per

inch, 4 ¾ turns will equal .1425”, and five turns will equal .150”.

These are performance instructions for aggressive cams and hard-line riding. Compare this with an average instruction of 2 ½ to 3-turns from most manufacturers.

Bob has one caveat with this approach. It is the standard correct methodology. The valve-spring seat pressure must match the camshaft. Weak valve springs may not control valve movement to that dictated by the rotating cam lobe, causing valve float and possible valve-to-piston contact. 18

I used my stock hydraulic lifters with these belt-gear drive cams and they worked fine all year long with aggressive riding.

So, there you have it, three methodologies for lifter

adjustment:

1. One using limiter kits,

2. The second following manufacturers' instructions,

3. The third, an aggressive adjustment for hard-line cams and spoiling-for-a-fight riding.

Each works well within its particular realm.

Clearance the Front Exhaust

Now, I will return to the belt drive installation.

- Install the ultra-cool belt guard and cover but do not torque.

- Install the front exhaust pipe loosely and gently rotate down until it interferes with the belt guard cover.

- Mark the pipe, remove, heat; compress the pipe with a rounded mandrel in the interference area gently and smoothly until there is a ⅛” clearance between pipe and belt guard.

- Install the complete belt cover, use anti-seize on the threads of the supplied four twelve-point stainless steel screws and torque to 30 in-lbs.

Have Some Fun

Install the spark plugs and follow Bob’s last instruction: “Kick the tires and light the fires, you’re ready to go. HAVE

SOME FUN!” 19

I can attest to the fun as I have been blasting around all summer on my Wood Knight Prowler (Silent) Advanced Belt Drive System.

1 S&S Cycle, Inc. Research and Development. Technical Support, Scott Sjovall; discussion

2 Davies, Jeff. E-mail. Attachment Hi, January 29, 2008, 9:59pm

3 Ibid.

4 Ibid.

5 Ibid

6 Ibid.

7 Ibid.

8 Ibid.

9 Wood Performance Carburetors. Web site, http://

(www.woodcarbs.com)/bdrive.htm Knight Prowler Belt Drive.

10 Andrews Products, Inc. Web site, http://

(www.andrewsproducts.com) Motorcycle Catalogue, page 20.

11 Wood Performance Carburetors. Web site, http://

(www.woodcarbs.com).”

12 Ibid.

13 Wood, Bob. Wood Performance Carburetors. E-mail, January 3, 2008 at 3:12 pm.

14 Wood Performance Carburetors. Web site, http:// (www.woodcarbs.com)/bdrive.htm Knight Prowler Belt Drive

15 Wood, Bob. Wood Performance Carburetors. Parts of a telephone conversation January 8, 2008, 10am., followed up with an explanatory e-mail.

16 Harley-Davidson. Service Bulletin, TT324, July 26, 2007. Twin Cam Crankshaft Inspection & Run-Out Specifications. Page 3.

17 S&S Cycle, Inc. S&S Catalogue 1958-2008, High Performance Parts for American V-twins Since 1958. Pages

9-26, Low Noise and Positive Valvetrain

18 Wood, Bob. Wood Performance Carburetors. Part of a telephone conversation January 8, 2008, noontime.

19 Wood Performance. Knight Prowler Belt Drive® Installation Page 8, Step 25.

# Chapter III

## H-D 110 Cubic Inch CVO

This chapter deals with so much more than the CVO 110 because the theory and practical information applies to common problems throughout all Harley-Davidson engines. This is the very reason I have spent so much time and effort on this chapter other than the fact I am blessed or is that more correctly cursed with literary diarrhea.

There is not much left that I find truly exciting after working on almost every Harley-Davidson engine over the last 35 years. However, the challenge to understand, effectively repair chronic problems and suggest design changes to treat

the causes instead of the symptoms is exhilarating for me.

If any of the following information is specific to only the CVO 110, I will let the reader know.

Again, the theory and practical applications will apply to similar situations in your engine no matter what it might be.

## I Had a Bad Feeling about Factory Big Inch Engines

Harley-Davidson's back is really up against the wall on this one. When the Motor Company began to build big-inch engines, I had a bad feeling. How could they go big and still stay within EPA and CARB guidelines on a platform based on two cylinders with the notoriously inefficient 45-degree cylinder angularity?

The only great things these cylinders and their firing order

have contributed to the

Harley world is the look and secondly, the potato-potato sounds from the exhausts. However, ever-quieter noise emission rules have muffled the sound almost to the point of extinction.

I feel the issues concerning the Harley-Davidson Custom Vehicle Operation's (CVO), 110 are all about heat, excessive heat.

Lower Compression Combats Heat but Robs Power

Part of the formulae for big inch engines is higher compression ratios. I will demonstrate the anemic 9.3:1 compression ratio of the CVO 110 as I proceed. Of course, I feel this is symptomatic of the Factory attempting to control

heat. In 2009, H-D further lowers CVO 110 compression to between 9.15:1 and 9.2:1 depending on model.

Lower compression runs cooler while higher compression brings heat to the equation. Moreover, leanness of the fuel mixture, in order to meet EPA dictates to reduce emission pollutants is the excess heat culprit.

Alternatively, is the real culprit Harley-Davidson's reliance on an antiquated design? Well, everyone knows that H-D's reliance on our beloved dinosaur technology bases on market demand at odds with a modern environmentally challenged world. The V-Twin design and technology bases on our, the riders wants and needs.

H-D's Problem with Meeting EPA Mandates

The Motor Company attempts to adapt a patchwork of different new or established technologies to an air-cooled engine that engineers felt would never run in the first place back in the early 1900's.

I will examine these pieced-together Band-Aids later in this chapter in the section titled: Solving the Tendonitis of Lean Burning Air such as EITMS. EITMS is the acronym for Engine Idle Temperature Management System. This feature turns off the rear-cylinder fuel injector at hot idle. The rear cylinder still functions without fuel. Air still sucks in on the intake stroke and goes through the remaining three strokes acting as air pump. The lower temperature air-cools the engine while only running on the front cylinder.

If meeting EPA dictates while going big with cooperative cylinder angles replacing the difficult Harley-Davidson 45-degree positioning like 56-degree or 60-degree angularity; more cylinders; a water jacket and overhead cams (OHC),

then there is good opportunity for successful compliance.

However, an air-cooled, 2-cylinder, pushrod operated relic with an awkward firing order at and does not have much chance of success as the years roll by. The firing order and positioning is within the 720-degrees of crankshaft rotation to complete one full cycle. The four theoretical strokes are intake, compression, power, and exhaust. I say theoretical because, in practice valves open and close before and after each stroke resulting in real-life extending and overlapping of the cycles.

The California Air Resources Board's (CARB), latest requirements were in 2008 while EPA regulations substantially tighten up in 2010. There are stringent automotive miles-per-gallon limits set for 2020 that are already modifying near-future model choices such as

eliminating some rear-wheel-drive cars because they are currently too heavy relative to front-wheel-drives. The future is coming fast and it will not be kind to tradition and nostalgia.

Increased cylinder angularity allows for a more efficient firing order and rhythm; more cylinders allow each to compensate for and bolster others through teamwork and overhead cams rid the engine of a bulky and inefficient valve train with mechanical limitations. Moreover, these directly actuating overhead cam lobes allow for an increased and responsive rpm range.

The water jacket and liquid cooling combat the effects of lean burning engine management systems. Water jackets bring under control the excess heat required to burn all of the fuel in a perfect world thus eliminating noxious pollutants. Furthermore, pre-ignition and the great engine destroyer, detonation has difficulty wreaking their havoc because the

precursor, heat lowers to manageable constant temperatures. Water jackets also address the second EPA anxiety, which is to suppress internal engine noise from the outside world.

## Design From the Past: the Nova and the 45c.i Flathead KL

Harley-Davidson has made many forays into functional engine development in the past including the developmet of the Nova by AMF in the mid-seventies. The AMF ultra secret Nova was a multi-cylinder, water-cooled, double overhead camshaft engine available in 6-displacements. There was a balance shaft for vibration reduction and an interchangeable 5-speed transmission and valve gears. The Nova scheduled introduction in 1981 collapsed in favor of the Evolution after the AMF buyout. The new owners lacked financing to lauch both ventures.

A progressive advancement for Harley-Davidson also buckled because of patent infringement issues with the famous Vincent Motorcycle Company.

In 1953, the year Indian closed its doors; H-D experimented with deviating from the inefficient angularity to an efficient angle with the KL.

The 1952 to 1956K-series were the flathead 45c.i predecessors to the Sportster.

I find information such as this and the Nova Project fascinating as their implementation would cause the transformation of the Motor Company into a flexible modern technological one decades ago.

Engine Statistics

The 110 cubic inch CVO Twin Cam motor uses the stock TC96 flywheels, which, as we know has a stroke of 4 ⅜" while TC88 engines have a stroke of four inches.

Both the TC96 and the TC88 have 3 ¾" stock bore cylinder diameters. If boring out the TC88 to 3 ⅞"; this will achieve a ninety-five cubic inch engine. Boring the TC96 to 3 ⅞" will result in 103 cubic inches.

Modifying a TC88 to achieve a 110 cubic inch displacement requires installing 4 ⅜" stroke flywheels, boring out the stock engine case cylinder spigots and fitment of four-inch piston-bore cylinders. Therefore, the 2007 CVO 110" engine as with earlier Twin Cam 110" engines use Twin Cam engine cases with four-inch piston-bore cylinders with a stock TC96 stroke of 4 ⅜". This is completely safe and reliable with all year Twin Cam engine casings.

At Heavy Duty Cycles, (www.heavydutycycles.com), we routinely build 116 and 124 cubic inch engines first introduced by S&S Cycle, Inc. using the stock Twin Cam casings with stellar results. Using a 4 ⅝" stroke flywheel assembly is still a safe way to achieve the 116" and 124" larger displacement engines. A 4.000" piston-bore with these 4 ⅝" flywheels result in 116.2 cubic inches. A 4 ⅛" piston-bore diameter with the 4 ⅝" stroke produces 123.6 cubes. Again, the engine case cylinder spigots need boring for fitment of these monster cylinders. I will explore these engines in Chapter XII: Cubic Inches: Strokers and Big Bore, in subsection, 124 Safe Cubic Inches.

Pertinent CVO Specifications

1 Measured at the engine; a percentage is lost in the powertrain as power transfers to the rear wheel. Engine torque is always higher than rear wheel torque by around 12%-15%. This is why state of tune of the motorcycle as a whole is so important. Torque transfer loss can be minimized by a correctly adjusted clutch, accurate alignment, properly inflated tires etc. or maladjustment of powertrain components can squander torque and horsepower before it reaches the tire patch on the asphalt.

2 Engine displacement formula is; X 0.7854 X stroke X number of cylinders. Most usually round off engine displacement for quick reference.

3 Cubic inch conversion to cubic centimeters: 1 cubic inch (c.i) = 16.39 cubic centimeters

4 The gasket surface flatness/warpage.

5 Runout measures on the shaft bearing surface by the flywheel. This measurement varies manual-to-manual and year-to-year. I would never install a set of flywheels unless they were .001” or less at the shafts.

6 This measurement also varies manual-to-manual and year-to-year. I would never install a set of flywheels unless they were .003” or less at the rim, preferably .001”. Some older Panhead and Knucklehead flywheels may be mismatched in size making these specifications unattainable unless replacing flywheel halves.

7 This is the automatic compression release (ACR), installation torque specification.

8 The lower figure is the minimum dry test compression; however, this figure is okay if wet test remains close. Squirt oil into cylinders (wet test), turn over a few times, and repeat compression test. If compression increases, piston rings are worn and not sealing correctly.

9 1.800” (45.7 mm) is the closed valve specification while 1.250” (31.8 mm) is the open valve specification. Subtract the two and we see the springs will apply 432 lbs at .550”valve lift.

## CVO 110 Sprocket/Pulley Sizes

TC96 and CVO 110 sprockets and pulley sizes are the same although dramatically different from before

## CVO 110 Transmission Gear Ratios

A transmission gear ratio indicates the number of transmission mainshaft (and attached clutch hub) revolutions to drive the final drive pulley one complete revolution. TC96 and CVO 110 gear ratios are the same. However, 2009 touring model gear ratios update.

## The Symptom: Blowing Rear Head Gasket

### The 110 Rear Cylinder Oil Leak

It appears that a number of 2007 CVO 110” engines are developing an oil leak on the backside of the rear cylinder just

below the exhaust port. Typically, this begins to occur just below two thousand to six thousand miles.

I have received reports of this malady occurring on the stock TC96 also although I have personally not seen one. Even if there are TC96 occurrences, I feel they will be far less of an issue than with the 110's. This will become evident as I proceed.

Even though head gasket failure does not appear to be a widespread difficulty with the TC96; both the TC96 and the CVO 110 are experiencing a flywheel scissoring problem, which I will be examining in Chapter V: subsections, Twin Cam Flywheel Misalignment and Left Crank Runout and Right Crank

The Email Alerts

I have been receiving many emails and queries from worried CVO 110 owners.

Jerry Rice from Memphis, Tennessee wrote, “Here is info on the problem I called you about today on my 2007 - Screamin’ Eagle Ultra with the 110” motor. Rear cylinder head gasket develops a leak typically around three to 5-thousand miles. Not a compression leak but an oil leak. (It) presents itself on backside of rear cylinder just below the exhaust port. Moco (Harley-Davidson Motor Company) has new head gasket (p/n 16801-07A), but this has not been a fix. Had mine replaced (along with the rear cylinder itself) at 9,000 miles first week of July…after 4,000 miles began to weep again. By time I got back home with 8,000 miles on the repair it was a pretty good leak with oil dripping down on top of primary case and entire back of rear cylinder and head with oil residue baked on. Also, seems that in the rear cylinder the steel insert is showing some degree of separation from the

aluminum cylinder. So Moco is replacing for a second time the rear cylinder and head gasket. Thought you guys might know something about this or be able to get the “straight scoop.” I personally know several folks with the problem and through my main Internet forum I have become acquainted with dozens more. This is NOT an isolated problem and seems to be one more of design than manufacturing problem. Thanks in advance for any advice / recommendations. If you have a fix, I’ll be glad to head up your way and get the work

A reader that wishes to remain anonymous from Marietta Georgia, wrote to the American Iron Magazine website AIMag.com) that stated in part, “I really do hope all of the recent activity on the AIM forum regarding the repetitive problems owners of the 110 inch motors are having will persuade the magazine to do some good solid investigative

reporting to dig up the true facts regarding the unacceptable number of bikes plagued with this problem. This problem is so severe that…the three other 2007 FLHTCUSE owners I know the best have all had this problem at least twice each. One is currently talking a new motor since his leak reappeared for the third time, one IS getting a new motor right now, but he's gone through two sets of replacement jugs trying to fix the problem and the third one called me last week to tell me he just found out that his bike, previously thought to have been fixed, is once again leaking. My bike is too leaking again after being "fixed" with the revised head gasket. I currently have the 16801-07A's installed (that replaced the 16801-07 gasket). The 16801-07A gaskets were new in April of this year (2007). That's a whole lot of gasket design to go through in 8 months if there isn't a real problem…Talk to several dealers. Talk to me. Talk to my friends that also have this problem. We'll give you VINs you can check with to see what's happened and when. This is a very serious problem that so far has gone on unnoticed in the press. Harley IS working on it,

but I think they need a little shove for them to get really serious about it. When we all went through the cam bearing saga in 1999 and 2000, the failure rate was a whole lot less and that problem got a whole lot more press time. EVERYONE was talking about that one. This time around, the failure rate is enormous and as of yet, nothing has been printed about it. Sure, the number of potential problems is a lot less this time around since it is primarily only CVOs, but the failure rate, even without including multiple failures, blows the cam-bearing problem away. 2

Another discouraged reader wrote the American Iron Magazine website AIMag.com) that also stated in part, “Unfortunately I have to write to you regarding a recent post in your AIM column. I have recently purchased a 2007 S.E. Springer with the 110 engine. I really love the bike and it had 1800 miles as of last week. I have heard all of the bad press on the 110 engine and was really hopeful that I would not

experience any problems. I noticed an oil mist around the rear head/ cylinder joint. When I took it to my local H-D dealer for a service I had them inspect it and they said the head gasket was leaking…The mechanic replaced it, took it for a test ride, and inspected again only to see that it was still leaking. This is what shocked me. He called some H-D tech line and was informed that yes, they are having issues with the 110 engines and he needed to pull both heads off for inspection of the liners. He pulled them off the next day and both liners had shifted (he showed me; there was a noticeable lip on both liners that your fingernail would catch on). Both cylinders are being ordered Monday along with new rings and a new gasket (he called it a 4th generation head gasket). While I understand H-DMoCo (Harley-Davidson Motor Company) not coming out with this info to the public, the shocking thing and what was frustrating to me was that it's not being communicated to the service network. My mechanic was very frustrated with that. 3

This same reader was very complimentary to his dealership and the concerned service he is receiving.

Donny's Comments on 110 Complaints

I have been investigating the reports of a CVO 110-rear cylinder leak since day one. However, Internet claims need verification. The writer may fervently believe their problem as widespread, but is it?

Is the problem diagnosis correct? If I cannot see it for myself or hear it from a known and respected source, then how can I write about it?

At Heavy Duty Cycles, we work on Twin Cams up to six days a week, year in and year out. I do get to see the problems first-hand, sometimes before anyone else. Some dealers currently profess (February 2008) not to have heard about the

problem at all.

Maybe they did not sell any because the CVO Screamin' Eagle models are the most expensive Harley-Davidson's and only available in limited quantities. Alternatively, maybe the ones they sold have no miles on them, which is certainly possible with some segments of the nouveau riding population.

However, after a deluge of commentary, I knew there had to be some legitimacy. I did not feel comfortable writing this article until I had personally seen a malfunctioning CVO 110, taken it apart, diagnosed the problem(s) and repaired them to my satisfaction.

One thing is clear; the Harley-Davidson approach is not working.

My approach will.

I went through the 1999-early 2000 cam-bearing debacle. It affected more bikes than this current malady because the rear cam ball bearing in the cam support plate was in every Twin Cam manufactured at the time.

The CVO 110 is a limited edition motorcycle and only available in certain models at that. I do not demean this problem situation in any way, as the percentage of bikes affected may well be higher.

The CVO 110 gasket leak seems to develop anywhere between 2000 to 6000 miles with a few occurring even earlier.

The early rear-cam-bearing problem could easily occur as

early as this but might also not develop for many more miles either.

As Jerry accurately says, the problem is one of an oil leak and not a compression leak. As for a replacement head gasket blowing again after a test ride, I can only say that the technician may not have checked for a warped gasket surface. I will explain later, the types of gaskets in 110 usages demand a very smooth surface in order to seal properly, which, in my experience is not always the case.

CVO 110 Rear Head Oil Leak

Pictured is the view from the exhaust port of the rear head and cylinder of an oil-leaking CVO 110 engine with removal of the rear exhaust pipe. The head and cylinder mate just

below the head's exhaust port. The finning expanse below the port is the top fin for the cylinder. The two studs with nuts attach the exhaust pipe to the head's exhaust port. Organized mechanics always rethread nuts to original parts like the pictured exhaust studs to prevent loss. The baked-in-oil stains are the otherwise polished cylinder fins below the exhaust port. The oil originates from the return oil passage that begins in the head cavity enclosed by the rocker arm wafers on top of the heads. The breather apparatus located between the two rocker arms in each head separates oil and air from the gaseous oil transfer, which flows up the inner pushrod tunnels and into the heads. The descending pistons forces air out of the bottom end compartment through the rollers of the pinion bearing into the timing chest and then up the hollow pushrod tunnels into the heads. The separated air vents out into the breather cover and then sucks back into the engine on the intake stroke. This separated oil returns to the timing chest by way of the continuous oil passage in the head, cylinder, and engine casing. The return oil must pass by and through two

sealed surfaces, the head gasket between the bottom of the head and the top of the cylinder and an O-ring between the bottom of the cylinder and the engine casing. Each head to cylinder and cylinder to engine casing juncture has a hollow dowel pin that protrudes up and down into the mating surfaces to ensure oil does not leak out. The base juncture uses an O-ring from 1999 to present around the hollow dowel to ensure further that the oil stays inside the passage. Interestingly, the use of the O-ring, part number 11273 also seals the dowel supported by a head gasket from 1999 circa 2003. H-D eliminates their use from 2003 to present. All I can say is that the use of an O-ring around the dowel pin works and we use them on all years, where possible at Heavy Duty Cycles. The oil leak begins as weepage from the dowel pin outwards. Compression still seals, as the gasket does not blow inwards.

# Disassembling the CVO 110: What I Found

The oil leak was evident as soon as Chris brought his CVO 110 in for repair on his own dime.

Many readers have complemented their dealer response as adequate in their emails and discussions with American Iron Magazine and They are not blaming the dealers as they can see their frustration.

However, Chris was not one of these. His out-of-town dealer had repaired his 110 twice already, insinuating that Chris had caused the oil leaking from the rear head gasket. Their third response was threatening to kick Chris out of the dealership for complaining too much. I mean, after all, he had about 5000 miles (8000 kilometers), on his machine. Why

should he expect more? I felt sorry for Chris but was excited to see and diagnose the problem(s) myself.

The Flywheels

Maybe, this CVO had the dual problem of oil leakage and scissored flywheels.

I will skip to the bottom end straightaway to deliver the good news in a bad-news situation. The 4 ⅜" stroke flywheels were in excellent shape. The pinion shaft runout was half-a-thou (.0005"). Chris's sprocket shaft measured in at one thou runout (.001), with both flywheel rims the same at one thousandths-of-an-inch runout (.001"). Chris has a nice flywheel assembly any mechanic would proudly install.

However, a nearby dealer has a CVO 110 in the shop with

a staggering 25-thousandths of an inch (.025”) runout on the pinion shaft. This occurs when the flywheel halves scissor, by shifting on the press fit crankpin, one in front of the other. Now, I do not yet know whether this scissoring problem is a result of the extra torque generated by the TC96 and CVO 110 over the TC88’s and their big bore counterpart the TC95 or if it is due to a relaxation of the .007” interference press fit straight crank pin into the flywheels halves. Perhaps it is a combination of the two.

Moreover, perhaps, the extra heat generated by the closed loop EFI Stoichiometric 14.7:1 air fuel ratio aggravates the press fit, expanding the flywheel body around the crank pin slightly. Certainly, there are reported air fuel ratios on the rear cylinder approaching a blistering and destructive 16:1 on the rear CVO 110-cylinder.

No matter, the Heavy Duty Cycles machinist Simply Marvelous tacked the crankpin with Tig welding to the

flywheel on Chris's CVO 110 power-side (sprocket shaft left side), using stainless steel rod just to be safe. He did the pinion right side also but one would think this not necessary since it only drives the oil pump and the valve train via the cams. Marvelous tack welded the crankpin equidistant on four sides in a crosshatch pattern.

Steve, my head mechanic for thirty years, ground the top of the welds. He was careful not to fool with their integrity. There is too much money involved to fool around.

The Cams

The SE-255 cams were also in excellent shape as was the cam support plate assembly and oil pump.

Chris, a gnarly and grizzled greybeard who has ridden his

whole life gave me one strict guideline to follow, “Fix the fuckin’ thing. No more problems, I bought this bike to ride.” When Chris cooled off, I suggested tongue-in-cheek that the real problem might be that he rode his bikes too much. Smiling, he told me to stick it where the sun don’t shine.

Well, one future problem may well be the ovaling of the cam-support-bearing parent-material-bearings. Moment Arm Force exacerbates with the 255 cams and the big-inch motor. I cover this extensively in Chapter II: Repairing the Achilles Heel, subsection, Moment Arm Torsional Bearing Wear.

This compounded by my desire to put a set of higher lift, longer duration with more overlap cams into the 110.

The 110 engines are begging the performance mechanic for the ability to release more potential horsepower and torque! Steve and I installed a 585G S&S gear drive cam

setup. With a stock rocker arm ratio (1.625), these cams have a .585” valve lift. However, we increased the rocker arm ratio to 1.675, effectively increasing valve lift to .603”.

Chris, now has a cam lift increase of .053” over the CVO 255 cams, which will give him some more bottom end oomph, overlap increases from a paltry to a respectable and duration increases on the intake cycle and on the exhaust cycle for better top end.

If this were my bike, I would have put in much larger cams but Chris is a high-miler. I am being conservative with his engine because of his bad experiences. Make no mistake though; after I revamp his engine it will be very much faster than a stock CVO 110; reliably fast.

The pistons we put in will increase the compression

marginally from 9.3:1 to 9.65:1.

## Comparing Stock H-D, Screamin' Eagle (S.E.), and the Cams in my Current Bike

Timing, duration, and overlap measure in degrees of crank rotation; the crank needs two complete revolutions totaling to complete the four basic cycles, intake, compression, power and exhaust. The camshaft rotates at half time of the crank rotation.

Valve-lift or simply lift and cam lift measure in thousandths-of-an-inch. In practice, the specification measurement of cam timing, lift, and duration begin at .053"

off the baseline heel of the cam lobe. Purists feel that the effect of this will compromise true overlap figures and analysis. Rocker arm ratio is stock Twin Cam 1.625. Stroked and/or big bore engines with more cubic inches can absorb more duration. The power band will also lower with bigger cube engines using longer duration cams. These are generalizations and we want more compression in tandem with a longer duration cam in big-inch engines. These factors are proportional and inter-related.

** BW-408B Knight Prowler Advanced Belt Drive System cams I am currently running in my 2007 103c.i Streetglide. Check out the wild differences with these cams compared to puny stock 2007 TC96 cams.

The Pushrods

Interestingly, the H-D CVO 110” engines use stock Twin Cam fixed length pushrods. One would think that Harley-Davidson would incorporate their Screamin’ Eagle adjustable pushrods into their Screamin’ Eagle engine. I put in the appropriate Screamin’ Eagle pushrods and pushrod tunnel kit (17997-99A), into Chris’s Screamin’ Eagle CVO 110 engine.

There was an improper installation of one of the middle pushrod O-rings during the previous attempt to repair Chris’s engine. The O-ring did not seat properly so oil leaked down and blew onto the head/cylinder fins and dripping off the back from there. Therefore, Chris had two oil leaks baking onto the CVO 110-cylinder fins and messing with components underneath.

The Rocker Arms

Here we go again! The Screamin’ Eagle engine also did not

use the Screamin' Eagle roller rocker arms but used stock rubbing rocker arms with pads. Performance 101 dictates the use of roller rocker arms with performance cams both for preventing frictional loss of horsepower and valve train longevity.

The upper-balled end of the pushrod fits into a mating pocket in the end of the rocker arm, which swivels up and down a distance equal to the cam lobe lift on a rocker arm shaft. The major mechanical reason for a rocker arm is it connects the pushrod on the opposite side of the head to where the valve is locates on the other side. The approximately 3 ⅝-inch long rocker arm transfers the dictates of the rotating cam lobe to the top portion of the valve stem encased in the single beehive valve spring, circa-2004 and later TC88 and TC96 or dual cylindrical springs, circa-2003 and earlier TC88's. The CVO 110 uses the dual valve springs with a dampening wave spring in between.

The stock rocker arm end-pad opposite to the pushrod side sits over the valve stem. The pad rubs along and down on the top of the angled valve stem, forcing it to open. This obviously creates friction both on top of the stem and on the sides as the pad pushes the valve stem against the angled valve guide; hence the frictional loss of horsepower and wear issues.

A roller rocker, rolls effortlessly on the top of the valve stem as it pushes the valve open with minimal side-force against the valve guide.

The function of the valve spring(s) is to close the valve back onto its valve seat after the elliptical cam lobe rotates past its highest point, compressing the springs. The springs push the valve closed as the valve train follows the descending cam lobe.

Now I can explain why the cam lift is less than the valve lift. If the rocker arm ratio were 1:1; that is, the extension of the rocker arm that cups over the upper balled-pushrod was on the same plane as the rocker arm extension on the other side that actuates the valve, then cam lift and valve lift would be equal or the same figure. This is the case with the 1936 to 1947 Knuckleheads. However, on every Big Twin from 1948 to present, one side of the rocker arm offsets to the other so a multiplication of the cam lift transfers to the valve stem thus increasing valve-opening lift.

The stock Twin Cam's rocker arm ratio is 1.625:1. Since I had to replace the stock rocker arms with roller ones, I took the opportunity to use those with an increased rocker arm ratio from the stock 1.625 to 1.675 to increase the valve lift (torque), relative to cam duration (horsepower).

Torque gets the beast going and horsepower keeps her going.

## Increasing Rocker Arm Ratio and Valve Lift

We put S&S 585G gear drive cams in the CVO 110 with 1.675 rocker arm ratio. Stock rocker arm ratio is 1.625 for all Evo and Twin Cams 1984 to present. Cam lift stays the same while valve lift increases from .585 lift to .603” lift when using a 1.675 rocker arm ratio. Stock S.E. 255 cams used in the CVO 110 engines has a lift of .550”

## Compression Ratio vs. Cam Duration

Stroked and/or big bore engines with more cubic inches can absorb more duration.

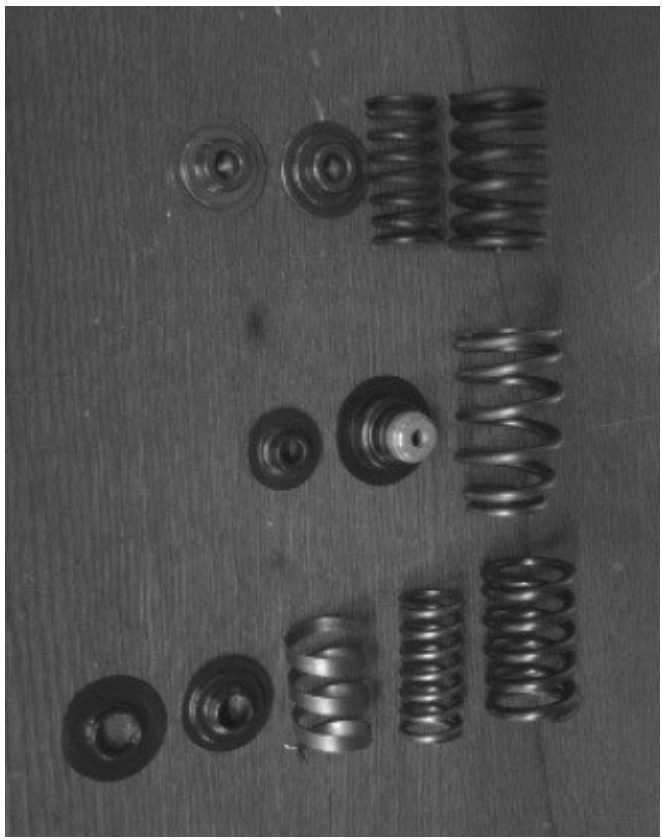
Cylinder pressure affects power. More cylinder pressure will usually increase horsepower. Three main factors determine cylinder pressure, which are compression ratio modified by camshaft duration and altitude.

Higher altitudes lower atmospheric pressure because there is not as much air. Therefore, less air results in less compression. As a general rule of thumb, the higher the altitude the less camshaft duration is necessary. This does not apply to visiting higher altitudes, as cam duration reduction is not possible without replacing the camshaft. Less duration is necessary when living and riding at higher altitudes most of the time.

Increasing cylinder pressure is determined by more compression and less cam duration. Conversely, a decreased cylinder pressure is the result of less compression and more cam duration.

Therefore, in performance applications more cam duration is necessary to bring in more air to burn more fuel, which creates more pressure to drive the piston down on the power stroke with increased force to turn the rear wheel with the following caveat. More cam duration will necessitate a higher compression ratio to maintain a higher cylinder pressure to maintain increased power.

Twin Cam Valve Springs



Pictured are the styles of Twin Cam valve springs for differing applications. The top set of dual cylindrical shaped valve springs (18201-83 outer and 18202-83 inner), sit, one inside the other and hold each valve closed on early Twin Cams with a stock .473" lift up until 2003. This spring set is one of the few parts carried over from the 1984-1999 Evolution motor into the Twin Cam engine. Valve spring design concentrates on both closing the valve when allowed by the rotational descending cam lobe and in controlling harmonics. We have all seen the pictures of a weaving and oscillating solid and fixed bridge in an earthquake. Well, valve springs, even though they compress and decompress sometimes surge or fibrillate with the rapid movements and demands of the rest of the valve train. These inertial and momentum forces sometimes compel the confused springs to move opposite to normal. This is why there is an inside and outside spring, each having different coil-pressure ratings. The coils-per-inch of each spring is also different to combat harmonics. If one-spring surges, the other can hopefully, counteract this. The

lower valve collar in the picture sits over the valve guide pressed into the head and is above the top valve collar, which has a center appendage that fits over the valve's stem. Two half-moon valve keys fit into a slot near the top of the valve stem, holding the top valve collar and the springs below to a predetermined height. The ridges on the collars are the respective seats of each spring. The middle picture is the beehive shaped single valve spring (18245-02) that operates the closing of each valve with .471"/.474" intake/exhaust lift up until 2007 and an increase to .484" lift in 2008. Its utilization began in some model Twin Cams in 2003. It is still in use to present day. The beehive spring combats harmonics by progressively changing coil distance and tapering the coils to the top of the spring. Both methodologies work as the coils are constantly changing in strength with progressive physical properties. The bottom valve collar incorporates a valve seal. When the seal wears, the whole collar must replace. The 1984-2003 dual springs fit over a separate valve seal pressed onto the top of the valve guide. The Screamin' Eagle beehive

springs (18013-03A), look the same but of course, allow more valve travel (lift), and have higher seat pressures; once again, to combat potentially increased harmonics with performance camshafts. The bottom picture shows the CVO 110 triple valve springs. They accommodate .550” lift cams with two traditional coil springs and an in-between flat wave spring. S&S Cycle has been using this methodology on its performance springs for years with great success. The wave spring interferes effectively with any developing untoward harmonics. The valve collars are thinner to help allow for the extra valve lift travel.

Upper Valve Spring Collar Clearancing

In any Harley engine, when using performance valve springs, it is necessary to check surrounding clearances. The spring coils may be a little wider. They may also be somewhat higher, especially when performance valves are also in use as with the CVO 110.

Performance valves not only have larger valve faces to promote the ingestion of increased amounts of air but they may also have longer stems to accommodate longer length valve spring packs, which give clearancing for higher lift cams.

In the case of the CVO 110, we worked on; all four upper valve collars were touching the inner lower-rocker-wafer. Furthermore, the rear intake collar was actually rubbing against the inner wafer. We look for a tolerance specification if between .030” and .060” clearance between the spring pack’s upper valve collar and the inner rocker wafer.

I could tell that the inner lower wafer had relief work done to it at the Factory, just not enough. The dealer had rebuilt this particular engine twice. Why had the dealer technician(s)

not checked this tolerance? We are back to Performance 101, which they probably do not teach in standard H-D PHD courses.

Anyway, there had been three kicks at the can. My main wrench, Steve properly clearanced the lower inner wafers with his Dremel high-speed grinding tool.

Twin Cam Valve Springs

Dual cylindrical shaped valve springs (18201-83 outer and 18202-83 inner), sit, one inside the other hold each valve closed on early Twin Cams with a stock .473” lift up until 2003. This spring set is one of the few parts carried over from the 1984-1999 Evolution motor into the Twin Cam engine.

The valve seal is separate from these spring sets. They press fit onto the tops of the valve guides.

Valve spring design concentrates on both closing the valve when permitted by the rotational descending cam lobe and in controlling harmonics. Valve springs, even though they seemingly only compress and decompress sometimes surge or fibrillate with the rapid movements and demands of the rest of the valve train. These inertial and momentum forces compel the confused springs to move opposite to normal. This is why there is an inside and outside spring, each having different coil pressure ratings that determine ultimate seat pressure. The coils-per-inch of each spring is also different to combat harmonics. If one-spring surges, the other's differing properties can hopefully, counteract these untoward movements.

Two half-moon valve keys fit into a slot near the top of the valve stem, holding the top valve collar and the springs below

to a predetermined height. The ridges on the collars are the respective seats of each spring.

An ovate beehive shaped; single valve spring (18245-02), operates the closing of each valve with .471”/.474” intake/ exhaust lift up until 2007 and an increase to .484” lift in 2008. Use of the beehive spring began in some model Twin Cams in 2003. It is still in use to present day. The beehive spring combats harmonics by progressively changing coil distance and tapering the coils to the top of the spring. Both methodologies work as the coils are constantly changing in strength with progressive physical properties.

The bottom valve collar incorporates a valve seal. When the seal wears, the whole collar must replace.

The Screamin’ Eagle beehive springs (18013-03A), look the

same but of course, allow more valve travel (lift), and have higher seat pressures; once again, to combat potentially increased harmonics with performance camshafts. Recommended valve lift maximum is .575”. Installed height of 1.800”, seat pressure is about 205-lbs force while maximum lift force increases to about 360-lbs.

The CVO 110 triple valve springs easily accommodate .550” lift cams with two traditional coil springs and an in-between flat wave spring. S&S Cycle has been using this methodology on its performance springs for years with great success. The wave spring interferes effectively with any developing untoward harmonics. The valve collars are thinner to help allow for the extra valve lift travel. They have superior metallurgy to compensate for their thinness.

Checking Valve Spring Travel for Higher Lift Cams

In addition, it was necessary to check the CVO spring pack to see if it would accommodate our newly installed .603” valve lift cam and rocker arm ratio combination. The CVO cams are .550” valve lift while stock TC96 2008 valve lift increases to .484” valve lift from earlier stock valve lifts of .471” on the intakes and .474” on the exhausts.

This simple measurement takes little time. It is not necessary to install the lower valve collars over their mating valve guide appendages that protrude up into the head unless the valve seal is part of the collar as on the circa 2003 and later beehive valve spring Always use a new valve seal during a rebuild; therefore, replace the lower valve collar with a new one in this particular beehive application. If the valve seal presses onto the top of the valve guide, replace with a new one. Then install upper valve collars with valve keys onto the valve stem with the valve closed on its seat, one valve at a time. It is necessary to hold the upper collar in place, keyed

with the valve stem. A Vernier caliper is a slide ruler with a gauge that measures in thousandths-of-an-inch. Measure the distance between underneath the upper color to the top of the valve guide seal.

Our measurements centered around .740", almost ¾-of-an-inch. Traditionally, we use a clearance tolerance of .060" but some manuals are now advising an increase to .080". If we deduct the tolerance clearance of .080", we can safely use a cam valve-lift up to .640". The manual says .650". However, it is safe to use .650" valve lift cams here, as there is lots of clearance. Always check tolerances with performance work, never taking any measurement or guideline for granted. In any case, our cams come well within specifications at .603" valve lift.

I sometimes find that the Screamin" Eagle catalogues can be confusing. Usually, the catalogues leave out key pieces of information that I need. In this case, it was difficult for me to

discern whether the CVO 110 engines utilize .575” lift or .650” valve lift spring packs. If .575”, which will accommodate the CVO SE255, .550” valve-lift cams, then our .603” valve-lift cams would certainly cause the upper collar to smash mercilessly into the top of the valve guide seal and perhaps the guide itself.

Why there are .650” lift valve spring packs in an engine with .550” valve lift cams is a mystery to me. The spring packs may be generic to a series of cams with differing and greater valve lifts although this is not the precise way to do this.

The reason is that the different valve lift cams require different valve-spring-seat-pressures. For example, a .575” valve spring pack may have an installed 175-lb seat force with 450-lb seat forces when compressed to accommodate the

maximum valve lift of the camshaft. Secondly, a .650” valve spring pack may have an installed 195-lb seat force with 500-lb seat force when compressed to accommodate the maximum valve lift of the camshaft. Do not use higher seat-force valve spring packs when unnecessary. The reason is simple. Why have the valves slammed into their mating valve seats harder than necessary? Harder valve return leads to faster wear and less valve life longevity. The performance mechanic always matches seat pressures to cam profiles. Less leads to possible immediate parts collision. Too much leads to premature unnecessary wear.

CVO 110 Rear Head Oil Leak, Head Removed

Pictured is a CVO 110 rear cylinder. Again, leaking oil stains the cylinder fins immediately behind the return oil passage. The oil-return-dowel-pin attempts to ensure the return oil

stays within the confines of the oil passage. The front dowel pin's function is for head-to-cylinder alignment. The rear dowel does double duty of alignment and oil return. Oil drains from inside the rocker wafers down through the head, cylinder, and engine case into the timing chest. The head gasket is the second-generation multi-layer-system (MLS) that H-D hoped would rectify the oil-leaking situation. This three-in-one gasket has a metal center with a rubber Viton coated metal gasket on either side. The two metal rings each on a gasket appendage opposite to each other on either side of the bore hold the multi-layer gasket together. This gasket will not tolerate a rough or warped gasket surface. Cross-hatching in the cylinder indicates that there are very few miles on this engine. Cylinders are fine-honed with cross-hatching to assist ring break-in and for oil retention on the cylinder walls. Above the cross-hatching is the area where the piston and rings change direction at top dead center (TDC) of the stroke. This piston-ring stop and reverse direction area wears more and faster because of the sudden change in piston direction.

Above this area in the cylinder is a small black band that is above the travel area of the top compression ring. This is carbon build up from the combustion process.

Valve pockets are visible on the CVO performance piston top to accommodate .550” valve lift cam. Stock TC88 and TC96 pistons are flat. Stock valve lift for 2008 TC96 is .484” with most TC88`s having a .473” lift.

Controlling Blowby (Oil Carryover)

There is no provision to control blowby in the CVO 110 except with the new oil pump, introduced on the 2006 Dynas and carried forward (26037-06) with some changes as reflected in a new part number for the CVO 110 oil pump. Oil carryover exacerbates with sustained higher rpm riding and/ or increased swept volume in the cylinders, which is a fancy

way of saying increased displacement or more cubic inches.

The reason is simple. The descending pistons displace an oilified mist up into the heads where oil is supposed to be separated from air by the breather apparatus' in between each rocker arm set, one in each head. The air then vents into the air cleaner and the separated oil flows via gravity down the oil return passage, again, one in each head. Sustained high rpm riding necessitates the pistons to move much faster and many times more per minute thus displacing much more oilified mist. Increased swept volume means that the pistons have greater diameter (big-bore), or move further up and down the cylinder bore (stroker), or a combination of both thus having greater swept volume displacing more oilified mist up into the rocker wafers.

Well, the CVO 110 has a standard stroke of 4 ⅜" used in the TC96 that is .375" longer than the 4.000" stroke of the TC88. However, the massive bore of 4.000" is a full .250"

larger than the stock TC88 or TC96 bores and still .125” bigger than the TC95 and TC103 engines respectively. Furthermore, Chris will ride at a good clip for long distances (sustained high rpm riding).

The CVO 110 oil pump (26277-06), as does the 2006 Dyna and TC96 oil pump (26037-06), removes much more oil from the timing chest and bottom end cavity back to the oil reservoir than the 1999-2006 oil pump series (26035-99, 26035-99A and 26035-99B). This helps control oil blowby in that there is not as much oil lying around to combine with displacing air into an oilified mist.

I asked Chris if he experienced any blowby situations. He said that, yes, a fine oily mist did vent to the outside of the engine but being an older rider with numerous oil carryover experiences did not view his 110 as having a problem worth

worrying about. Chris is a tolerant guy as long as his bike does not stop running.

I asked Steve to install an S&S Cycle, Inc. (www.sscycle.com) reed valve, which fits between the oil pump and the inner right hand engine case. The reed valve controls displacing air that travels through the pinion bearing rollers from the bottom end cavity into the timing chest where it normally continues up the pushrod tunnels into the head rocker wafers. I discuss the reed valve at length in Volume I: The Twin Cam, Chapter IV: Harley Oil Leaks: Marking its subsection, The S&S “Reed Valve” Solution.

Steve then enlarged the oil drain hole in the heads to .103” to assist oil drainage out of the heads from the stock TC88 size of .060” and the previous upgrade recommendation of .089”. Last year, in my 2007 TC103, I became the guinea pig for trying a .110” drain hole.

It worked for me so I asked Steve to do the same for Chris making the already enlarged drain hole even larger. It is easy to make this hole bigger but very difficult to make smaller if going too far. I also discuss this enlargement technique in further detail in the same chapter as above in Volume I: The Twin Cam

I then decided to put this blowby issue to rest for good. I asked Steve to replace the stock single-chamber-oil-separator H-D breather apparatus with a Doherty Machine (www.dohertymachine.com), dual-chamber-oil-separator Mystfree breather in each head to separate more of the oilified more efficiently. The Mystfree has other design attributes that further assist in oil separation, which I discuss in Chapter IV in Volume I: The Twin Cam.

I could have just as well replaced the rocker wafers with those from TP Engineering (www.tpengineering.com), that also incorporate a different type of dual-chamber-oil-separator from that of the Mystfree, again with other separation features that are a study in the Physical Laws of Nature (Physics). Once more, please see Chapter IV in Volume I under the subsection, The TP Pro Vent Solution.

Other causes of oil blowby are usually mechanical installation errors inside the timing chest. These errors originate with a misaligned oil pump or pinched, wrong size or missing O-rings between components from where the oil distributes to other engine compartments. Steve does not make these types of mistakes during reassembly.

Now, I am going to get into the controversial stuff, head gaskets and supposedly separating cylinder liners.

You will not believe this One: Stock TC96 EFI on the CVO 110

I was looking for bigger fuel injectors for the EFI throttle body to get a little more gas into the cylinders to lower the air fuel ratio (AFR), below 14.7:1. I was looking at different sized throttle bodies also.

H-D, in the Screamin' Eagle 2008 catalogue has a 58-millimeter throttle body with matching injectors. Ahh, I thought, it is perhaps too big for the CVO 110. I wanted something smaller.

Reading further, this unit is for engines with more than 100 horsepower at the rear wheel. Realistically, without changing a bunch more stuff, I figured Chris‵ bike would have somewhere between 100 and 110 rear wheel horsepower when done. I still thought this throttle too big.

One of the first lessons in Performance 101 is that bigger is not necessarily better. Installing too big cams, jets, carburetors, EFI throttle bodies, injectors, valves; you name it can easily decrease torque and horsepower.

My 2007 TC103 pushes over a hundred horsepower and it runs great with the upgraded 50mm throttle body and matched injectors. The stock 2007 46mm stock throttle body is bigger than what went before.

My choices from Harley-Davidson were injectors from the 50mm throttle body or those from the 58mm one. Okay, I reasoned, the H-D EFI is an automotive Delphi system. I was going to go to a GM dealership and then the performance speed car-shops looking for bigger injectors.

I asked Steve to get the part numbers off the CVO 110 EFI throttle body and the two injectors, one to feed each cylinder. As he wrote down the part numbers of both throttle bodies and injectors he exclaimed, “Boss they are the same as yours!”` The 2007 CVO 110-throttle body is 27708-06 and the injectors are 27709-06A. My 2007 TC96 throttle body is 27708-06 and the injectors are 27709-06A.

The 2008 CVO 110-part number changes to 27605-08 because of the servomotor that is integral with the throttle body for the wire-by-throttle control.

The 2008 injectors are 27609-01B, which indicate a change from the injectors in use since 2001 to 2005, part number 27609-01A. I think we are going backwards again.

Now, let us go back to the situation at hand, “Okay, Steve, so the 2007 CVO 110 has the 50mm injectors and body?” “No,

boss, they are identical to your stock TC96 46mm throttle body!" The 50mm throttle body part number is 27623-05 and the injectors are 27654-06. I thought Steve was nuts so we both checked and he was right as usual. I made a wrong assumption. I assumed the throttle body and injectors were bigger than the 46mm used on a 2007 stock TC96 engine.

I was exasperated! Why would they do this; to save a few pennies like with the pushrods and rubbing rocker arms? No, I thought, this has to do with emission restrictions. Well, at least I had my answer.

I went into a 2007 Screamin' Eagle catalogue and ordered a 50mm throttle body with matching injectors.

The 50mm throttle body and injectors are part of my Band-Aid strategy to assist the two major fixes to resolving

Chris' CVO 110 over-heating problem, the cause of the symptomatic blowing head gasket. These will only work on the 2007 CVO but not on the fly-by-wire 2008 to present models that require a servomotor to activate the throttle plate.

Gas cools combustion temperatures by lowering the AFR ratio.

Each Band-Aid is not a solution in itself but further insurance Chris will have no more over heating problems. Other Band-Aids will be the addition of the Harley-Davidson cylinder head fan (part # 91550-00C, chrome; 91531-00D, black) and a one-quart larger oil tank, both to combat excessive heat.

I could not use 2008 catalogue because the only upgrade is the monster 58mm for racing only body, whose part number

is 27623-05 and the injectors are 29915-08. In my opinion, the 2008 catalogue is much thinner from resultant emissions control crackdowns.

## CVO 110-Cylinders and Liner Sleeves

Okay, here we go, this is the biggie, or is it?

From what I gather, most are assuming that the head gasket problem on the CVO 110 rear head emanates from a shifting, separating or moving cylinder liner. Liner separation seems to be the favorite description.

### Separate Cylinder Liners

Many liners or sleeves, especially in the old days were a

separate part from the cylinder itself. When worn, the mechanic simply presses the sleeve out of the cylinder and presses a new one back in. The new liner would require boring or at least honing to crosshatch the cylinder walls to assist in ring break-in. Boring the new liner was preferable to eliminate bore tapering and out-of-round conditions. Honing, unless with a sophisticated and expensive tool called a Sunnen hone, will only follow existing bore out-of-round and taper conditions.

This is the most expensive tool, I purchased for my shop, Heavy Duty Cycles over the years. I saved and waited for five years before a good deal came up at an auction back in the early eighties. I still had to finance it. Then it took a few years more to purchase all the special size mandrels and hones, to precision fit every Harley-Davidson bushing and race. I still have it today.

The 45-degree crosshatching also serves for oil retention in

the cylinder wall to assist in piston-to-wall lubrication.

Spiny Lock Cylinder Liner

Twin Cam cylinders have a spiny lock iron sleeve that casts into the high-pressure die cast cylinder. The top of the cylinder sleeve is part of the head gasket surface. Machining the alloy cylinder and its cast-in liner flat for the head gasket surfacing occurs at the same time so one cannot be higher or lower than the other can. If so, why can I hook my fingernail on the liner on a used rear CVO 110-cylinder?

I am going to use a TC 88 cylinder for comparative purposes throughout my examination of the CVO 110. The TC88 cylinders has been in use for nine years now with nary a problem in many hundreds of thousands, if not millions of Harley-Davidson's. Nine years, did the Motor Company not

cease production sooner in 2007 with the advent of the TC96? No, nine years because both the TC88 and TC96 cylinders are the same with a 3 ¾” bore with the same height of 4.930” and a cylinder base flange diameter of

However, the pistons are different because the TC96 pistons redesign for lowering their reciprocating weight and they are shorter to clear the ⅜”-increase in stroke.

Pressing Out the CVO 110-Cylinder Sleeve

The rear sleeve visually looks to be separating and every indication is that its cause is a problem with heat.

When the Motor Company manufactures its cylinders, never mind the year or model, one of the final machining steps is to plane the top gasket deck surface perfectly flat for excellent gasket sealing capabilities. This machining

procedure assumes the cast iron cylinder liner and the aluminum cylinder body are at one with each other. The Factory planes both to an equal flat gasket surface.

The front CVO 110-cylinder sleeves that I have seen do not visually seem to be separating and they pass the fingernail-hooking test. However, the cylinders, like all Twin Cam cylinders are identical and are interchangeable, front-to-back and back-to-front.

In fact, when I run my fingernail across the top cylinder deck gasket surface on a used and disassembled CVO 110 front cylinder, from outside in towards the cylinder bore it is a smooth progression. My nail does not hook or detect sudden changes in gasket surface height. The front cylinder, because it is in the airstream rarely broaches the heat prostration that the shrouded rear cylinder endures. Therefore, front cylinder

liner separation is not an overt problem with Stoichiometric air-fuel-ratio of 14.7:1 or mercifully less; that is a richer and cooler condition.

Moreover, would a TC88 cylinder suffer the same separation fate with newly instituted 14.7:1 closed loop 14.7:1 air fuel ratios? Well, the TC96 cylinder is the same as the TC88 cylinder! The TC96 also must reach Stoichiometric closed loop AFR on both cylinders. Obviously, its rear cylinder shrouds also and must suffer higher temperatures. There is no widespread problem with the rear TC96 cylinders suffering the same negative heat effects that plague the CVO 110's.

After all, the heat generating compression ratios are almost identical with only a .1 differential; the TC96 is 9.2:1 while the 2007-2008 CVO 110 is 9.3:1.

The clues are becoming more apparent. There must be a

difference between the CVO cylinder construction and that of the TC88/TC96 ones.

Now I run my fingernail across the top cylinder-deck gasket surface on a used and disassembled CVO 110-rear cylinder from outside in towards the cylinder bore, it is not a smooth progression. My fingernail hooks on the edge of the cast iron cylinder sleeve that is supposed to be flat and on the same level as the aluminum part of the top cylinder-deck gasket surface.

I did the same on every TC88/TC96 used cylinder in my store of which, there are many. Not once, did my fingernail hook on any of them. This is another clue that points to the construction of CVO 110-cylinders and the TC88/TC96 cylinders. They must be different.

A little further on, I am going to hack out the cylinder liners to look at their internal construction. I can now say definitively that the TC88/TC96 cylinder will not suffer from a separating sleeve.

Many are beginning to assume the cylinder sleeve is a separate part the cylinder and not spiny-lock cast in.

Pressing Out the CVO 110-Cylinder Liner

First, Steve heated up the rear CVO 110-cylinder, while on the press with resultant oil weepage from between the liner and the cylinder. At this point, even Steve and I felt there might be some merit to a moving cylinder liner. However, the engine is too modern to have a separate cylinder sleeve. This is a throwaway society and Harley-Davidson is no exception as fewer actual repairs occur in favor of parts replacement. Replacement liners are for the rebuilders of yesteryear.

Spiny lock cylinder construction adds rigidity and support. This is one of the main tenets of the Twin Cam design philosophy.

However, the oil did weep from between the liner and the cast alloy-cylinder body. I will discuss much more on how this is possible further on.

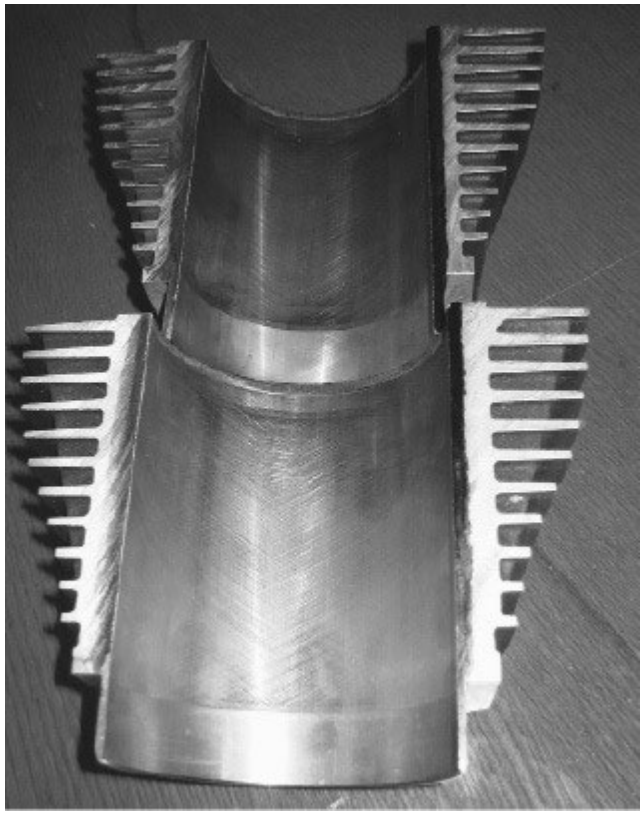
Steve began the attempt to press out the CVO 110-rear cylinder liner with our 20-ton capacity press. It got scary. The press creaked and groaned under maximum effort. Steve put a longer bar on to activate the press so he could move off to the side further from harm's way. He was afraid the cylinder was going to shatter under the force applied. Finally, the press fixture began to bend. Steve had to stop; certain the cylinder was going to shatter. The cylinder sleeve did not move one-

thousandths of an inch.

We now know that the liner is not a separate part. Yet it still fails the fingernail test.

Let me get a few more red herrings out of the way before I ask the Heavy Duty Cycles machinist Marvelous Martin to cut a rear CVO 110-cylinder in half and do the same to a TC88/ TC96 cylinder for comparison purposes. We will be able to observe first hand, the actual cylinder-liner construction in minute detail.

Cross Section TC88 and CVO 110 Cylinders



Pictured are the cross-sections of a TC88 cylinder on top and a CVO 110 problematic cylinder underneath. The difference in bore sizes is evident; TC88 has a bore diameter of 3 ¾" while the massive bore of the CVO 110 increases to 4.000" a full ¼-inch or .250" bigger in diameter. The cylinder sleeve is not press fit into the cast portions of the cylinder. In fact, on both the TC 88 and CVO 110-cylinders, the cast iron sleeves are spiny lock cast into the aluminum cylinder bodies. Therefore, we cannot remove the sleeve nor can it move up or down in the cylinder. The left side spiny locking is the way it appears naturally. On the right side spiny lock construction, I took a black magic marker and crudely delineated the sleeve from the main cylinder body. The question becomes; if the liner cannot move then why does there appear to be separation of the liner from the cylinder body at the top gasket deck? A fingernail rub across the gasket surface towards the cylinder bore hooking on a ridge right where liner and cylinder body meet evidences this! Furthermore, oil seeps out between

sleeve and body when subjected to heat. The 45-degree crosshatching in the cylinder bores indicates that neither cylinder has much mileage. The crosshatching assists the rings in their break-in period. The angularity is necessary for this function. Rings would not break in correctly if the fine abrasions were on the same horizontal plane. The crosshatch also serves for oil retention in the cylinder wall to assist in piston-to-wall lubrication. From top to bottom on both cylinders, there is a narrow circumference band of carbon buildup. Combustion byproduct carbon burns on in this area. It locates above the furthest distance travelled by the top compression piston ring. The next area is wider. It is wear. This is where the rings stop at the top, (TDC), of the cylinder on their stroke and change direction. The crosshatched section is the largest area of the cylinder where the pistons and rings travel. The next narrow circumferential band is the area where the rings once again stop and change direction at the bottom of the stroke (BDC). Finally, the last portion is the sleeve flange that extends down into the motor case spigot

hole. The flange's dual function is first; the external outside portion receives and gives symbiotic support with the engine casing. The second is internally, where its extension gives support to the piston near and at the bottom of its stroke. Remember the piston extends further down after the lowest oil ring set. Both cylinders have eleven cooling fins. I measured them to see if both proffered the same cooling capability. It is an optical illusion that this picture makes the CVO 110 finning look greater than the TC888. In fact, the measurements of the TC88 offer much more finning area. The TC88 has eleven fins, the same as the CVO 110. However, the length of the finning on the TC88 is approximately .130" (⅛") to about .165" greater all way round the circumference of the cylinder and from top to bottom. Therefore, the CVO 110 has a substantial reduction in fin area, as there is less length in its finning all the way around and all the way from top to bottom. A ⅛" to extra area of finning extrapolates into a substantial difference in total area of finning between the two barrels. Furthermore, this area differential greatly affects the

cooling capabilities of the two cylinders, the adequately cooled TC88 cylinder, and the inadequately cooled CVO 110-cylinder. One might logically think that this situation would be the opposite with the CVO 110 having more area finning than the smaller and cooling running TC88! Why would the Factory reduce finning area on a hotter running cylinder? Perhaps its rationale revolves around marketing the look of the V-Twin engine so important to the Harley rider. An increase in finning area to at least match that of the TC88 and hopefully go beyond with even more area to dissipate extra heat that is such a problem with the CVO 110 rear cylinder in combination with the .250" wider bore would produce a much fatter top end. The cylinder head fins would need to increase to be symmetrical with those of the cylinders, which is a good thing because as we will see the rear CVO 110 head gasket area warps with the extreme heat. In my mind, this scenario is wonderful; just look at the extra cooling capabilities that result. Of course, the other reason for fin-area reduction might coincide with cost reduction or at least cost

savings. Now, we have yet another compelling reason for a CVO 110-cylinder redesign. However, the situation becomes worse with the CVO 110 oil return passage being in too close proximity to the cylinder bore. This, in combination with the cylinder studs being closer also to the bore and not sitting in the middle of the gasketing surface further compounds the aggravating factors necessitating a cylinder redesign. These are some of the reasons I am using an aftermarket cylinders on the CVO 110's I repair in conjunction with oil cooling, a cylinder head fan and lowering AFR air to fuel

Is the Liner too thin?

I have received many emails complaining the crux of the problem is that the CVO 110-cylinder sleeves are too thin. A typical one relates the following theory-explanation. "There seems to be a lot of questions regarding issues with the 2007-2008 Harley's. Regardless of the engine issue, most of the problems trace back to the lean Air Fuel Ratio of 14.7:1.

Face it, the rear cylinder gets the most heat so while the front can marginally survive the rear is expanding and contracting more and with a thinner sleeve for a 110", where is the mystery?" Some knowledgeable websites proffer the same explanation. If the liner is too thin, why does not the front cylinder at least exhibit milder symptoms than the identical rear cylinder?

Well, I guess no one took the trouble to measure the liners of TC88, TC95' TC96 and TC103 cylinders to compare with the CVO 110. My measurement for the CVO 110 actually came out less than what the readers email stated. His measurement is 110" while mine is 100". Different tools and margins of error can account for this as .010" differential, which is not a big deal for our comparisons. However, a margin of error this big would definitely cause problems in more precise applications.

I took the following measurements at the cylinder base or

bottom where the liner-flange exits the barrel.

- The TC88 cylinder 3 ¾” bore has a liner thickness of approximately .162”.

- The TC96 cylinder 3 ¾” bore has a liner thickness of approximately .162”.

- The TC103 3 ⅞” bore has a liner thickness of approximately .100”.

- The TC95 cylinder 3 ⅞” bore has a liner thickness of approximately .100”.

- Finally, the TC110-cylinder 4.00” bore has a liner thickness of approximately .100”.

Well, I guess the measurements take care of the “too thin” argument as the TC95, TC103 and TC110 liners all measure the same + or - .005”. The TC95 and TC103 liners do not pose untoward problems that I have observed…yet. The TC95’s are defunct because they are bored out versions of the discontinued TC88 engine.

These engines never had to operate under the Stoichiometric air-fuel-ratio. The TC103’s may encounter heating problems with the rear cylinder if no fuel modification were undertaken with the big bore TC96. Why would anyone big bore a TC96 into a TC103 without installing an ignition/fuel modifier? My current two favorites for EFI are the Terminal Velocity and the venerable Power Commander. The TC96 and thus the TC103 do not use carburetion; if so, I would include the S&S IST (Intelligent Spark Technology), ignition as one of my favorites. I will discuss these ignitions and others at length in Chapter IX: Fuel

Delivery, subsection, Ignition and Fuel

I do not think that the liner thickness is a factor for the CVO 110 problems although it could easily be made thicker in a cylinder redesign if it made some people feel better.

The Thrust of the Descending Rear Piston

This theory, regarding a combination of CVO 110-stroke and connecting rod length causing a destructive piston thrust force on the rear cylinder wall almost did not make it to this article, as it is ridiculous. However, I do not want it to survive so let's blast it. I can discount this bogus theory immediately.

Pistons, whether in the front or rear of all Harley-Davidson engines thrust against the cylinder walls particularly when changing axis (direction). Yes, it is a legitimate argument to

replace the awkward with a more friendly or 60-degree design but this is not the argument at hand. However, every other Harley-Davidson cylinder including the Knuckle, Pan, Shovel, Ironhead, and Evo survived piston thrust.

The Twin Cam cylinder is shorter than all the aforementioned barrels. Additionally, it is at one with its high-pressure cast in liner, in spiny lock construction. I would venture to say the Twin Cam cylinders are the strongest and most rigid of any in the history of Harley-Davidson. Furthermore, the ever growing use of synthetic oil reduces the thrust pressure as does the hypereutectic piston material and the Teflon coating on its thrust surfaces; enough said!

The Cylinders are not a Plug Fit in the Engine Cases

Normally, the lower exposed cylinder sleeves known, as

flanges fit tightly into the engine cases otherwise known as a plug fit. The aluminum engine cases expand more and at a greater rate than the cast iron sleeve appendages so there is never an issue of the tightness squeezing the cylinder and distorting it. The plug fit adds the support of the engine case to the cylinders.

It surprised Steve and I to find that the outer diameter of the barrel flanges were about 4.180” but the spigot hole in the engine cases are 4.208”. This leaves .028” for some potential shifting.

The cylinders sit over and hold down by four equidistant cylinder studs. Once the male-threaded cylinder studs that protrude from the base decks of the engine casings up through the cylinders and the heads tighten down to specifications via threaded, hollow female bolts; it is impossible for different component parts to move when cold.

As engine heating and cooling occurs, thermal expansion and contraction of different components may induce some movement for a few reasons. The non-plug fit of the cylinders give the room if there is pressure on the parts to shift while expanding and contracting. The cylinder and head loose fitment over the four cylinder studs also leaves room and may not discourage some little movements. Metal expansion even for alloys is different because of diverse metallurgical content of various aluminum alloys, (cylinders, and engine casings). The thickness of the alloy engine casings is greater than that of the cylinders giving subtle differences in expansion-contraction rates. The cast iron cylinder liner will expand at a different rate again from the casings and cylinder bodies.

Steve, Marvelous, our machinist and I stared at the cylinders some more with combined Harley-Davidson

mechanical experiences totaling over 120 years. There it was…another clue. The cylinder liner is thicker in the cast in spiny locked cylinder than when the cylinder flange protrudes from the base of the cylinder.

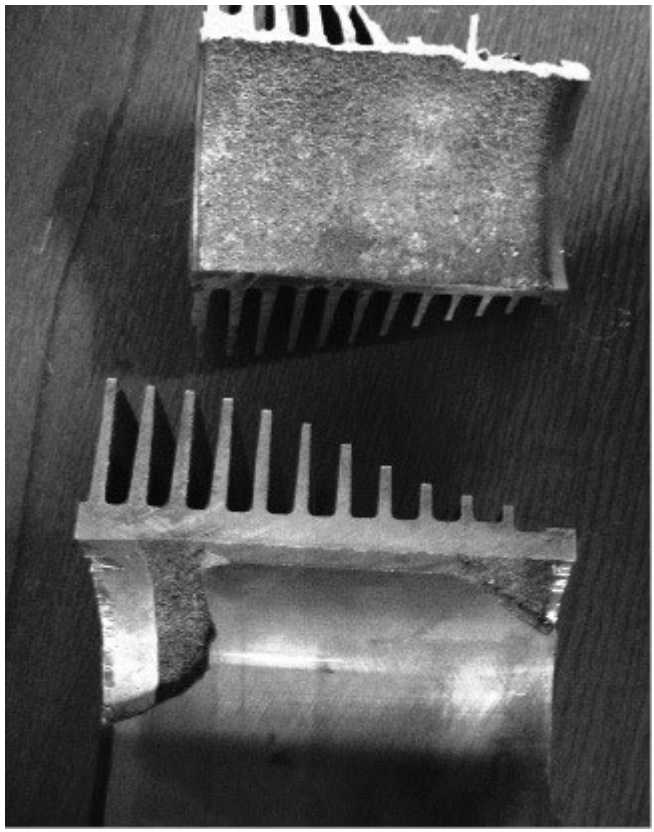
Steve postulated that this thicker portion of the sleeve must sit on the engine case spigot decking to add support for the cylinder liner lip's potential shifting. For this to occur, the cylinder must be a plug fit into the engine case spigot holes. The CVO 110 does not offer this support with a total of .028" extra room, which is .014" on either side.

The theory became that the liner can shift downwards from the force of piston-descending thrust and from metal expansion without engine case support. In other words the cylinder liner can However, if there is a spiny lock cast in cylinder liner, this still could not happen. In addition, if this situation occurs in the problematic rear cylinder, why would it not repeat in the front one?

These clues turned out to be red herrings.

At Heavy Duty, we have seen many bored out engine casing spigots for bigger bore cylinders returned to stock displacement usage. While this is certainly not ideal, one must appreciate how difficult it is to make a round hole smaller. The point, I am making is that I have observed many of these situations over the years and no appreciable shifting occurred.

Removal of Cylinder Liner Portions, TC88 and CVO 110



Pictured are two cross sections Twin Cam cylinders with component parts hacked away. The bottom cylinder is part of a CVO 110 rear cylinder, with a portion of the cylinder liner chiseled away top (left), and bottom (right) to expose the spiny lock construction of liner and cylinder body. Forgive the chisel marks, as we did not know what we would find underneath the cast in liner. Assisting us was our newfound experience at wrecking perfectly good parts in the destruction of the top TC88 cylinder, whose liner (not shown) we have torn away completely. The TC88 cylinder exposes the spiny lock construction of the cast in liner and cylinder body. Here, we now have the answer to the shifting cylinder and liner separation that many but not me blame for the rear head gasket leak. The TC88 head gaskets have never leaked oil once in my extensive experience. It seem that a high percentage of CVO 110's that are ridden do experience rear cylinder head gasket leaks. The TC88 liner is spiny lock cast in with the cylinder body right until near the juncture of the gasket surface. The space where the spiny lock ends is visible

by the shiny portion of the cylinder beside the grainy spiny lock material. This portion extends down to about the beginning of the top cylinder fin. It cannot move or shift. However, the CVO 110 liner is not spiny locked in for its full length. The CVO 110-cylinder liner is spiny locked cast in with the alloy cylinder body up to within .625” of the top cylinder head gasket surface! The other Twin Cam cylinders have cast iron sleeves that are spiny locked cast in with the cylinder alloy up to a distance of .094” from the top head-gasket cylinder deck! We now have the reason for a shifty cylinder liner on the top cylinder deck gasket-surface of the CVO 110. The cast iron cylinder sleeve dimensions not spiny locked with the alloy cylinder body at the bottom of the cylinders just above the protruding cylinder flanges are really a red herring. The TC88 dimension is .250” while the CVO 110 is .156” Any expansion or movement has no effect on gasketing (cylinder O-ring), between the engine case and the bottom deck of the cylinder or the piston support since this area is below ring travel. The top portion of the liner does not

cast integrally but mates with the cylinder body via very fine shallow ribbing, so fine that it is not visible in this picture. It is a large area extending down from the top cylinder deck to about where the second cylinder fin ends. This area can move under extreme heat with the differing expansion rates of the cast iron liner and the aluminum alloy-cylinder body. This is an aggravating factor to gasket failure. However, the head gaskets do not fail in this area. There are no dangerous compression leaks from the combustion chamber outwards. The gasket fails on the other side of the dowel pin far away from the cylinder liner as evidenced by the annoying oil leak. The shifting cylinder liner should not occur and will necessitate a cylinder redesign. In my opinion, the rear cylinder head gasket oil leaks occur because of extreme heat generated by air fuel ratios that may be as high as 16.0:1 in the shrouded CVO 110 rear cylinder that is supposed to run at 14.7:1 AFR in closed loop EFI mode according to EPA dictates. This extreme heat warps the head's gasket surface from inside to out, still sealing in the compression but

allowing oil to escape from a dowel pin that has no O-ring to assist sealing. So many areas need redesign here to cope with the Stoichiometric AFR heat generation. The moral of this story is that in building big inch engines, any cost saving shortcut can lead to ruin. Big inches do not tolerate all that a stock engine will.

Less Cooling Fin Area on the CVO 110

The TC88 has eleven fins, the same as the CVO 110.

However, the length of the finning on the TC88 is approximately .130” (⅛”), to about .165” greater all way round the circumference of the cylinder and from top to bottom. Therefore, the CVO 110 has a substantial reduction in fin area, as there is less length in its finning all the way around and all the way from top to bottom. A one-eighth-of-an-inch to extra area of finning extrapolates into a substantial

difference in total area of finning between the two barrels.

The extra bore diameter of the CVO 110 requires more finning to circumnavigate the bigger cylinder but it still has less fin area than the TC88 barrel. Furthermore, this total fin area differential greatly affects the cooling capabilities of the two cylinders, the adequately cooled TC88 cylinder, and the inadequately cooled CVO 110-cylinder.

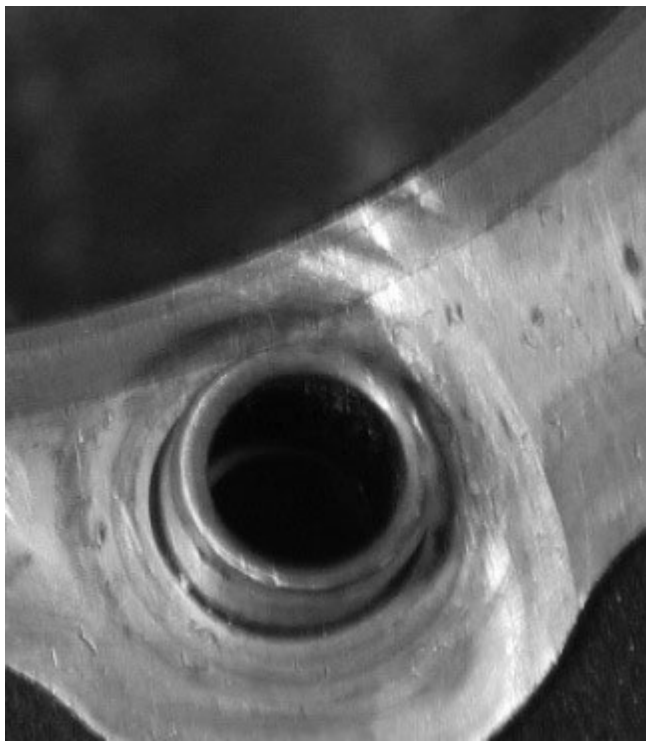
A thinking person would logically think that this situation should reverse with the CVO 110 having more area finning than the smaller and cooler running TC88! Why would the Factory reduce finning area on a hotter running cylinder? I think it has to do with marketing the look of the V-Twin engine so important to the Harley rider. An increase in finning area to at least match that of the TC88 and hopefully go beyond with even more area to dissipate extra heat that is

such a problem with the CVO 110 rear cylinder in combination with the .250” wider bore would produce a much fatter top end. The cylinder head fins would need to increase to be symmetrical with those of the cylinders.

In my mind, this scenario is wonderful; just look at the extra cooling capabilities that result. Of course, the other reason for fin-area reduction might coincide with cost reduction or at least cost savings. Now, we have another sensible reason for a CVO 110-cylinder redesign.

However, the situation becomes worse with the TC 110 oil return passage. It is closer to the cylinder bore than the other cylinders mentioned.

CVO 110 Rear Cylinder Oil Return Passage Dowel



Pictured is the CVO 110-rear cylinder oil-return passage dowel in the cylinder top deck head-gasket surface. The dowel

does double duty as a cylinder-to-head alignment-locating pin and as an oil router. Evo engines suffered weeping head gaskets because the oil return hole mated on the head gasket surface between cylinder and head without O-ringing or a dowel routing assist. At Heavy Duty Cycles, we have been machining problematic Evo weeping head gasket surfaces by machining a larger hole in both head and cylinder to accommodate a hollow oil-routing dowel pin for two decades now. Then we use a Viton rubber O-ring around the dowel as an extra sealing measure. The Twin Cam instituted this sealing system much to their credit in the TC88 from 1999 to circa 2003. However, H-D discontinues the dowel O-ring after then thus eliminating this sealing design. Going from bottom right to the top left of the picture; we begin with a wrinkle black powder coated cylinder fin. Then we move into the lighter grey head gasket surface. The oil-return dowel pin locater resides in this head gasket area. Then the darker grey cylinder liner that is spiny lock cast in with the cylinder body surrounds the shadowed cylinder bore. On the far side of the

dowel pin, we can see that it is just inside the cylinder sleeve. Look at the markings of the first gasket used on the CVO 110's and its upgraded version. This engine is on its third rear cylinder head gasket oil leak repair. The cast iron liner and the alloy cylinder body expand and contract at different rates. I can run my fingernail along the alloy gasket surface towards the cylinder bore and it will hook onto the edge of the cylinder sleeve. The multi-layer-system (MLS), gasket design depends on a flat surface to seal in compression and to seal out oil. This cylinder liner lip in conjunction with a warped head gasket surface wreaks havoc with MLS gasket sealing capability. Compounding this problem is there is not enough space between the dowel and the cylinder liner. The Cylinder needs a redesign for a variety of undeniable reasons. The cylinder and head finning is inadequate, less than a TC88 finning area and needs to increase in surface area for heat dissipation. Both the dowels and the cylinder studs need relocation to the center of the gasket surface. Then one consistent metal type will surround the dowels. The cylinder

studs will be able to apply more even pressure to seal the head gasket. Replacing the four cylinder studs with five equidistant ones will ably assist in combating the exceptional rear cylinder heat. The alloy composition of the heads also needs changing to resist high temperature induced warpage. Finally, the spiny lock methodology in use with the TC 88-cylinder needs reintroduction into the CVO 110 spiny lock design. However, not all the above solutions may be necessary if there were a reduction in combustion chamber temperatures evolving from above Stoichiometric 14.7:1 air fuel ratios.

Cylinder Liner Interfering with Dowel Pin Hole

I postulate that cylinder liner thickness is not a compelling factor in the TC110 rear gasket failure.

I have not seen or heard of a rear CVO 110 head gasket

blow in this area of compression sealing; that is unless the cast iron liner pushes up the gasket, still sealing the compression but causing a problem with level sealing as the gasket encounters the alloy. The rear CVO 110 gasket seals on the inside compression area but the gasket blows outwards from the centrally located oil return dowel. I realize that I am going back and forth on some subjects but all of this is part of my thinking process as I continue to search for the final answer, which will come.

It appears that all the Factory Twin Cam engine cylinder sizes use the same cylinder stud pattern, with the same distance between each. The gaskets for all the above bores have the same stud-hole locations but the CVO 110-oil passage is closer to the bore than that of the TC88. I am sure this has legitimacy and becomes a symptom of the cause.

The problem, thus far is that the Motor Company is addressing a symptom instead of the cause(s).

Steve and my inspections of the various Twin Cam cylinders result in some interesting observations and theories as we have been staring at them for weeks now.

- The TC88 inside liner is about .110” away from the dowel-pin oil return hole.

- The TC95 inside liner is about .90” away from the dowel-pin oil return hole.

- The TC96 inside liner is about .110” away from the dowel-pin oil return hole.

- The TC103 inside liner is about .90” away from the dowel-pin oil return hole.

- The CVO 110 inside liner is a paltry .030” away from the dowel-pin oil return hole.

A moving cylinder sleeve might well interfere with the dowel-pin-hole return oil passage causing a leak especially if the head warps from heat. How can the sleeve move though?

The CVO 110-cylinder liner is virtually pressing very close on the dowel pin outer diameter. Heat expansion of the hotter liner may distort the dowel pin-hole and disrupt gasketing.

It is becoming readily apparent that a CVO 110-cylinder redesign is necessary in conjunction with dealing with excess heat from air fuel ratios well above 14.7:1.

Cross-section View: Cutting the Cylinders in Half

As my Ma used to say, “The proof is in the pudding”. I am not sure what that exactly means but cutting a known excellent cylinder like the TC88 barrel in half to view the cross-sectional construction and compare it to a similarly cut problematic rear CVO TC110 will show the problem first hand debunking some theories and lending credence to others.

The first thing, our machinist, Marvelous Martin noticed is that the TC88 sleeve is a different cast iron alloy composition Marvelous had to cut the TC 88 cylinder in half at a much slower pace with less intense cutting. The CVO 110-cylinder cut much more easily as the iron alloy is softer.

As I say elsewhere, the TC88 sleeve is about .030” thicker on each side for a total of about .060” than CVO sleeve. Thus, the TC88 cylinder would take more effort to cut because of the thickness. However, it is still a harder material.

The second observation is obvious, the construction of both cylinders looks to be the same. Both cylinders are cast spiny lock construction.

No wonder the CVO 110-cylinder liner refused to budge in my 20-ton hydraulic press. There was a contest going on between the two, whether the press would break first or the cylinder would shatter. Twenty tons is a great deal of pressure and indicates just how strong and rigid these Twin Cam cylinders really are.

The cylinder wall thickness is also a red herring because a TC88 cylinder with a .162” thick bored out to a TC95 configuration leaves a cylinder liner thickness of approximately .100”, which is the same as the CVO 110.

TC88, TC95, TC96, TC103, and CVO 110-cylinder Studs

Steve and I began to measure the distance between cylinder studs on the various Twin Cam engines. The TC88 side-to-side (left to right facing front of the bike), cylinder stud measurements are 3.720” apart. Front to rear, the distance is 3.120” Therefore, the TC88 studs are 3.720” apart from side-to-side and 3.120” apart from front to rear.

Now we did the measurements for the TC96 engine and found them to be the same as the TC88 engine, which is not a great surprise. Obviously, the cylinder stud distances will be the same for big bore TC95 and TC103 engines as they’re smaller bore TC88 and TC96 counterparts.

The CVO 110 measurements are 3.720” apart from side-to-side and 3.120” apart from front to rear. Wow, the Factory is

using the same cylinder stud dimensions across .250” bore differentials from the TC88 to the TC110!

Now, how can the studs on both engines be in the middle of the gasket surface to apply equal pressure? This is one more redesign that needs to occur.

Although, in fairness I must say that the S&S 116”, 4.000” bore cylinders and the 124”, even bigger 4 ⅛” cylinders use the stock Twin Cam cylinders stud dimensions when building up these engines with stock motor casings. I think the differences in success ratios exist in the better construction of the S&S big bore cylinders and the fact that their engines run cooler than the CVO 110 because of their favored use of an IST ignition system.

The IST has an anti-knock component, which means it senses combustion chamber detonation (spark knock), and

changes ignition timing to eliminate the knock. Detonation occurs from heat so the IST ignition allows sufficient fuel and altered timing to combat the effects of excess heat.

## The Missing Clue

Something was still bothering me. There was a missing clue. My TC103 has the same liner thickness as the CVO 110. It subjects to similar heat in the rear cylinder, perhaps more because of my increasing the compression ratio to 10.5:1 much more than a CVO 110 9.3:1 compression ratio. Actually, not more heat because of the installation of my dual Jagg 10-core oil coolers and some fuel modification. However, it is still a very hot running engine. I rode my bike long and hard in very hot weather and yet I have no problems.

Why is the cylinder liner in the CVO 110 shifting or

moving and it does not occur in all the other Harley-Davidson Twin Cam cylinders, S&S cylinders, JIMS cylinders…need I go on? Ahaa, the missing clue; there is something more to see in the cross sections of the two cylinders. I asked Steve to smash out part of the liner at the top and bottom of the cross sections of both the TC88 and the problematic CVO 110-cylinders. The answer lay before us on the workbench. The thickness of these pieces were not really the clue even though the CVO 110 liner at the top of the cylinder deck is .060" less than the equivalent TC 88 measurement but again when bored out to a TC95 we are back at square one; the liner thickness is not the problem.

In any case, both thicknesses of the TC88 and the CVO 110 are equal at the bottom of the liner just before the flange. The following measurements are what I found.

- The thickness of broken piece of TC110 top liner is 150".

- The thickness of broken piece of TC110 bottom liner is .160”.

- The thickness of broken piece of TC88 top liner is 210”.

- Finally, the thickness of broken piece of TC88 bottom liner is .160”.

The Reason for CVO 110-cylinder Liner Movements

The smashed out top liner pieces gave me the answer I was looking for. I would never have found the cause had I only cut the cylinders in half. The reader will now realize why there are so many theories floating around.

No one has looked underneath the spiny lock construction! The CVO 110-cylinder liner is spiny locked cast in with the alloy cylinder body up to within .625” of the top cylinder head gasket surface! The other Twin Cam cylinders have cast iron sleeves that are spiny locked cast in with the cylinder alloy up to a distance of .094” from the top head gasket cylinder-deck! We now have the reason for a shifty cylinder liner on the top cylinder-deck gasket surface of the CVO 110.

The .625” area at the top of the CVO 110-cylinder attempts to hold the liner rigid and at one with the cylinder body via fine striations circling the inner diameter of the liner and the outer diameter of the cylinder body that mate and ostensibly prevents liner movement. These striations prove unsuccessful with the hotter rear cylinder. Under normal heat, this methodology would probably work as evidenced by the front cylinder where the liner does not shift under normal operating conditions.

The cast iron cylinder sleeve dimensions not spiny locked with the alloy cylinder body at the bottom of the cylinders just above the protruding cylinder flanges are really a red herring. The TC88 dimension is .250” while the CVO 110 is .156” Any expansion or movement has no effect on gasketing (cylinder and dowel pin O-rings), between the engine case and the bottom deck of the cylinder or the piston support since this area is below ring travel.

Cylinder Liner Comparative Dimensions

These measurements are taken from actual cylinders and are + or - .005”.

Measurements will vary from cylinder to cylinder. Therefore, these figures are for comparative purposes for analysis of differentiation or sameness and the clues that evolve. I measured the cylinder-liner thickness measurements from the external flange at the bottom of the cylinder. The width or thickness of the internal top liner and the internal bottom liner is from pieces my head wrench, Steve broke off on both a TC88 and a CVO 110-cylinder. The cylinder cast iron sleeve on the CVO 110 is only spiny locked cast in with the cylinder alloy body up to a distance of .625" (⅝"), from the top head gasket cylinder-deck! The other Twin Cam cylinders have cast iron sleeves that are spiny locked cast in with the cylinder alloy up to a distance of .094" from the top head gasket cylinder-deck! We now have the reason for a shifty cylinder liner on the top cylinder-deck gasket surface of the CVO 110. The cast iron cylinder sleeve dimensions not spiny locked with the alloy cylinder body at the bottom of the cylinders just above the protruding cylinder flanges are really a red herring. The TC88 dimension is .250" while the CVO

110 is .156” Any expansion or movement has no effect on gasketing (cylinder O-ring), between the engine case and the bottom deck of the cylinder or the piston support since this area is below ring travel. Axtell measurements are from what I can only see, as I did not cut one in half to remove pieces of the liner. The flange length of the Axtell cylinder is much longer than H-D cylinders providing more rigidity, support, and stability for both the cylinder and its piston. The piston at BDC must stop, change axis and direction. The piston needs all the support it can get in this area with all that it must contend.

## Why is just the Rear Head Gasket Blowing?

I cannot believe the Factory had a part in the following. Therefore, the dealer-mechanic that rebuilt the engine the last time used a handheld sanding disk to grind the head gasket surface to clean it up. He or she was trying to solve a slight warpage problem the wrong way. Their actions did not

resolve the warpage so it continued to add to the problem.

At Heavy Duty Cycles, we did this the proper way, precisely on a milling machine.

Why is just the rear cylinder blowing the head gasket? Why is the head gasket just blowing from the oil passage outwards? These two questions need analytical answering!

The rear cylinder is hotter than the front. We have known this forever, as it is a situation, which exists onwards right from the beginning of the V-Twin.

CVO 110-cylinder-Head Gasket Surface Warps from the Center to the Sides

Marvelous planed .007” off the rear head gasket surface in order to make it flat. The front head is okay but we equalized it with the rear and took .007” off it also.

The manual says to replace the head if warpage is .006” or more. The modern way to restore components is to replace them. Planing works exceptionally well, restoring parts to original working condition.

The warping occurs across the two rear head ports (intake and exhaust), close to the dowels so this could be the cause of the oil leak. The rear head warps up from the center to the sides. The head gasket surface warps on the rear head only. The front cylinder head could possibly warp also from the excess heat of a mistimed engine. However, this is not normal on a properly timed motor.

Improper Heat Treating

Improper heat-treating at the Factory could possibly be a cause for head warpage under high temperatures especially in the shrouded rear cylinder head where the air fuel ratio goes above Stoichiometric anywhere from 14.8:1 well into the fifteens with some reports going to an AFR of 16.0:1. If this were an annealing problem, where heat-treating removes gases to stabilize the alloy material to ensure the aluminum cannot shift under operating temperatures, it would also happen to the front head.

Symptoms and the Causes

Now, I will need to intersect the symptoms of head warpage with the shifty top cylinder liner to arrive at a conclusion. We cannot fix the symptomatic problems until we know how to fix the cause. Fixing symptoms will give longer

engine life but not long enough, as the root cause(s) will return repeatedly.

Cylinder head warpage and a shifty cylinder liner are mere symptoms.

Potential fixes will eliminate both through redesign of metallurgy to combat warpage, better tie-down of the mating parts and spiny lock cylinder liner to cylinder body construction to very near the top of the CVO 110 liner. However, these redesigns are attacking symptoms only.

For example, another symptom-repair would be to use a thicker fibrous gasket like that of the TC88 to take up gasket surface roughness and minor warpage. The CVO gasket is a newer technology that requires a flat smooth surface to function correctly. It is also thinner than the TC88 counterpart is.

One cause of the shifty top liner, the head warpage, and the rear head-gasket oil leak is excess heat with parts not built strong enough to endure the effects.

Now, the first question becomes what is the source of the excess heat. Secondly, why is excess heat not occurring in earlier engines but only in 2007 and later engines?

## EPA Perfection: Stoichiometric Air Fuel Mixture

Two relatively new phenomena relate to punishing demands for an air-cooled engine by the Environmental Protection Agency (EPA). They are Stoichiometric air fuel ratios (AFR), and closed loop electronic fuel injection (EFI).

I suppose the acronymists will designate OLEFI and CLEFI to represent open loop and closed loop EFI.

## Forgive the Acronyms

Forgive me if I keep showing and using acronyms with their long form definitions. I do this, because nothing is quite as frustrating as reading an article where the author assumes that everybody knows what the acronyms stand for. If the reader does not know the long-form meaning of one crucial acronym, then how is he or she supposed to comprehend the thoughts portrayed?

## Stoichiometric Air Fuel Mixture

The magic number for EPA guidelines and thus the ratio during closed loop operation of an EFI is 14.7. Everything in the engine management system including the sensors,

electronic control module (ECM or ECU), and electronic fuel injection revolves around this Stoichiometric number of 14.7 parts air per one part fuel (gas).

The Stoichiometric Ratio is the optimum air-fuel-ratio (AFR), where all the available oxygen in the air combines with all the available fuel for burning in the combustion chamber.

Stoichiometric AFR is different for various fuels; Gasoline is 14.7; diesel is 14.6; Methanol is 6.4 while ethanol is 9.0.

Predecessor engines to the Twin Cam had less cooling fin area and contained more existing heat inside the engine. They also ran a richer AFR, below today's closed loop EFI mandated Stoichiometric AFR of 14.7:1.

How do we burn such a lean mixture of fuel? Heat is the short answer. Cylinder head design and mating piston dome shape assist mightily but the bottom line is that spark plug gap increases to allow a hotter spark to ignite the thinned out mixture. The hotter spark generates from a higher voltage coil within the sophisticated engine management system. Gasoline cools but there is not much in a Stoichiometric mixture.

How does someone come up with a name like Stoichiometric? Learning becomes difficult because the defining word is intimidating never mind bizarre.Well, I assumed the fellow that figured all this out in the distant past somewhere lent his name to the concept. As it turns out, stoichio derives from the Greek word stoikheion, which roughly translates into element. The metric part of Stoichiometric sounds very familiar and identifiable as a classification of measurement. In a convoluted way, the answer comes from metriā, then metron and from there on, into the word measure. Therefore, Stoichiometric roughly

translated means measure element

It follows that the Stoichiometric Ratio of the element(s) in air relative to gasoline is 14.7:1.

The subject gets more complicated but understandable when explained properly. Stoichiometry is no fly-by-night theoretical mumble-jumbo as it rests upon three incontrovertible laws of physics;

1. The Law of Conservation of

2. The Law of Definite Proportions also known as the Law of Constant Composition,

3. And the Law of Multiple Proportions.

Mercifully, I will spare the reader from further discussion although I am sure the word Stoichiometric no longer seems so alien.

One last point, to drive the subject home since Stoichiometry will become a mechanical household word in the future is that;

- chemical reactions combine in definite ratios of chemicals.

- Chemical reactions can neither create nor destroy matter.

- Furthermore, elements cannot change into another element like the old scam of changing lead into gold.

- Next, the quantity of each element has to be the same throughout the overall equation or reaction.

To use a familiar example, the gases hydrogen (H2), and oxygen (O2) combine to form the liquid, water with two hydrogen molecules bonding with one oxygen molecule, 2 H2 + O2 = H2O.

Lambda

Lambda is the actual air fuel ratio divided by the Stoichiometric Ratio for that particular fuel.

$$\text{Lambda} = \text{actual AFR} \div \text{Stoichiometric AFR}$$

Example 1: if the actual air fuel ratio is 14.7:1.

$$\text{Lambda} = 14.7 \div 14.7$$

$$\text{Lambda} = 1$$

In the case of gasoline with an EPA desired fuel ratio of 14.7 to one divided by the Stoichiometric Ratio for gasoline of 14.7 equals a Lambda of one (1). Therefore, if Lambda is one (1), then the AFR is equal to Stoichiometric.

Example 2: if the actual air fuel ratio is 14.3:1.

Lambda = 14.3 ÷ 14.7

Lambda = .973

A rich running engine that by definition has an AFR ratio less than Stoichiometric has a Lambda less than one.

Example 3: if the actual air fuel ratio is 15.2:1.

Lambda = 15.2 ÷ 14.7

Lambda = 1.034

Conversely, a lean running engine that by definition has an AFR ratio greater than Stoichiometric has a Lambda greater

than one.

We will get maximum HP and torque with a Lambda slightly less than one. A Lambda of less than one translates into Stoichiometric under 14.7 parts air to fuel, which is no surprise.

Two Types of EFI

There are two main types of electronic fuel injection to monitor the various sensor readings such as ambient temperature, engine temperature; throttle position; manifold air pressure (vacuum), crank position and others.

Once a stock component part replaces with another like an upgraded exhaust, high flow air cleaner or longer duration-high lift cams, these changes will disturb the factory-installed

air/fuel calibration settings that closely link with ignition tables in the electronic control module (ECM). In order for a Harley to run properly with a performance change or changes, adjustments are necessary to give the ECM obligatory instructions of how much air and fuel are required. This is not so simple. Carburetors require mechanical jetting changes, which is easy. EFI needs electronic adjustments fed into either the ECM or modifying signals after they leave the ECM. The ECM fuel and coupled ignition tables are fixed.

Up until very recently from 1995 to 2006, Harley-Davidson as well as most of the aftermarket chose open loop EFI for the ECM to receive, interpret and control fuel settings from sensor readings. The vast majority of aftermarket ECM modification systems to alter fuel settings are very limited to specific changes without further adjustment. They cannot adjust to new components on their own. Therefore, the Factory or the aftermarket calibration tables will no longer be valid when making a change such as an exhaust system.

Sometimes, this will be injurious to the engine if a lean condition develops. Other times the fuel will become too rich and wasteful. It is possible to rewrite the stock ECM calibration tables but hardly practical.

Most aftermarket units allow a choice of settings or modified tables for specific changes but they need changing every time if making another modification.

The automotive industry has been using closed loop EFI forever favoring it over open loop EFI. Harley-Davidson finally made the switch over to closed loop operational fuel control in 2006 with the Dyna series. It carries over into the full 2007 lineup including Sportsters.

The reason is the addition of one of the now more

important sensors, the O2 or oxygen sensor. This sensor monitors oxygen content in the exhaust stream out the exhaust system. It is real time monitoring and diagnosis of the air-fuel-ratio's efficiency. Now the fuel delivery can correct automatically to the Factory calibration tables basing on EPA dictates.

Whether one agrees with the preset calibration tables or not, the point is that they become accurate in the predetermined and thus correct air-fuel-ratio.

Many aftermarket exhaust companies have begun the switchover by supplying their exhausts with threaded openings known as bungs for optional oxygen sensors. This allows easy installation of O2 sensor(s).

Self-tuning ignitions like the Terminal Velocity can now

recalibrate over 500 times per second to adjust to not only any changes the rider may make like an air cleaner but to constantly changing load conditions that occur while riding like adding a passenger or trailer, going up hills, going down hills, accelerating, decelerating or simply cruising. The ignition no longer cares as it simply reads the exhaust stream, analyzes the parts-per-million oxygen content, and delivers constant information in milliseconds to the ECM. The aftermarket ignition will make a decision, again in milliseconds to determine fuel requirements instantly. It will also compensate for altitude changes, lowering fuel as elevation increases because of less available air, and increasing fuel as elevation approaches sea level for the same reason of now more air.

This is new to the motorcycle industry but again, not the automotive especially in racing circles. Some pretty big names use this technology such as BMW, Mercedes, Honda, and the domestic Big Three. Furthermore, Formula One racing also

uses this expertise.

## Oxygen Sensors

We need some more gas both for cooling the combustion chamber as well as power.

Harley-Davidson uses narrow band oxygen sensors that work well in both stock and most performance applications. As I have said, O2 sensors measure oxygen content in the exhaust pipes and send back signals to the ECU (ECM) brain for analysis to reconfigure fuel and relative air delivery into the combustion chambers.

The exception is with supercharging and turbocharging applications, two examples of forced induction, where wideband oxygen sensors are necessary to deal with higher

boosts and rapidly changing rpm Forced induction compresses air above atmospheric pressure without a vacuum necessary for natural aspiration thus increasing the intake air-charge. Compression forces more air into the engine, providing a With the additional air in the boost, more fuel can inject into the charge, and the power of the engine increases substantially and almost immediately. Lambda AFR can change too fast for narrow band sensors to respond adequately, perhaps leading to engine damage. This is why faster reacting-wideband O2 sensors are a good idea in these instances. Lower boosts of 6 to 10-lbs can still use the narrow band O2 sensors without dangerous spikes in air leanness.

Closed Loop Electronic Fuel Injection

A closed loop EFI requires oxygen sensor(s) to feed exhaust composition Lambda to the electronic control module. Closed loop EFI functions at Stoichiometric air-fuel-ratio of 14.7:1. Closed loop needs a full circle to reach its mandated AFR. The

full circle ends with the ECM determining pulse width of the fuel injectors, which is how long they stay on. This decides the amount of fuel injected.

The engine goes through its strokes until completing the exhaust cycle. The O2 sensors measure Lambda and send a signal back to the ECM. The ECM will increase, or more likely in the real world decrease fuel injector pulse width to reach a Lambda of one (1), which is equivalent to Stoichiometric AFR of 14.7:1.

Closed loop EFI functions in cruising situations where fuel requirements lend toward a regulated Stoichiometric 14.7:1 AFR as well as hot idle.

Open loop functions for accelerating and cold idling particularly during engine warming after start-up.

Open Loop EFI

It is unlikely that open loop EFI will reach an AFR of 14.7:1. However, the likelihood is a richer condition of around 12.0:1 AFR with a Lambda of less than one (12.0:1 ÷ 14.7:1 = .816 Lambda), will exist with open loop. An air fuel ratio of 12.0:1 is the H-D engine default rich idle mixture.

When the computer (ECM), decides the engine has warmed sufficiently it will put the idle into closed loop and seek a 14.7:1 air fuel ratio. Blipping the throttle or otherwise taking the engine past 1200 rpm from idle of 1000 + or – 50 rpm will engage the closed loop mode and heat up the engine further.

Open loop EFI is not EPA functional for the cruising air fuel mixture ratios on the newer bikes when closed loop

14.7:1 AFR comes into effect.

Accelerating and rich idling cannot use the lean AFR used in Stoichiometric closed loop EFI because the engine will stall at idle and stumble while accelerating generating even hotter injurious cylinder temperatures.

These operating conditions require a richer AFR.

In closing, our current new Harley's in 2006 and later use open loop EFI during start-up with initial rich idling. Once engine temperature warms up the idle, the ECM will enter closed loop mode and seek Stoichiometric 14.7:1 AFR. Open loop EFI is required for accelerating to cruise mode and for passing because of a richer AFR requirement.

The ECU utilizes closed loop EFI while cruising to meet

EPA mandates.

It is not an ideal situation for an air-cooled engine.

## The Problem is Oil Seepage not a Compression Leak

As I have said, the CVO 110 rear gasket leak develops typically anywhere from about 1800 miles on. The problem is one of an oil leak and not a compression leak, which is an odd situation for a head gasket.

Usually, when a head gasket compromises in any way, a compression leak will follow. Most times, a compression leak will destroy the head gasket integrity from the inside out. I have not heard of any CVO 110 rear-head-gasket leaks developing into a full-blown head gasket although, I think it would develop and happen if left unattended.

All the cases I have heard of began early in the warranty period. Therefore, the struggling dealers who at this point do not have adequate information or the fixes are dealing with them all in early stages of leakage. Parts replacement will not solve this problem. This is all the dealers have now to attack the problem outside of revamped gaskets that cannot smother the symptoms, let alone attack the cause.

## Compression Leaks

A compression leak is a serious mechanical problem that needs addressing immediately. Compression leaks result from a blown head gasket from the combustion chamber into the atmosphere where red-hot gases escape from the engine. These gases act like a blowtorch and if left unattended, will destroy the gasket surface of both the head and cylinder creating a progressively larger pathway by melting

surrounding metal.

Oil Leak

The head-gasket oil leak originates from the return-oil-passage outwards through, above, or below a compromised head gasket. An affected rear head gasket will typically begin to weep an oil mist. The weeping progresses into an actual oil leak with more mileage and then oil dripping down on top of the primary case and the entire back of the rear cylinder and head with resultant baked on oil residue that will drive any new H-D owner nuts.

Oil will travel up or forward because of wind eddies at speed. In addition, the wind and gravity will also push and pull the oil down and back, as one would expect.

The gasket from the return passage inwards remains sealed

and contains the cylinder's burning fuel and compression as designed.

Gaskets

The factory uses an O-ring, part number 11273 on 1999 to circa 2003, depending on model, Twin Cams to seal the oil inside the downward return oil passage. For whatever reason, I do not understand, the factory discontinued the 11273 O-ring. This 11273 O-ring, installs around the hollow dowel that fits up into the head oil return passage and down into the cylinder return hole. It prevents an oil leak every time.

The hollow dowel acts both as a locating pin for mating head and cylinder and as an oil leak preventive measure. At Heavy Duty Cycles, we still use 11273 O-rings on all of our performance work where possible as well as normal rebuilds

on TC88's and TC96's.

It is not possible to use an O-ring on the TC95 or TC103's because both utilize a stock 3 ¾" TC88 or TC96 sized-cylinder that bores out to 3 ⅞" to gain 95 cubic inches and 103 cubic inches respectively. When the cylinders are bored to 3 ⅞" there is not enough room for the O-ring on the compression side of the dowel. The same situation exists with the CVO 110 4.000" bore.

Another clue exists here. Why do the 95ci and 103ci cylinders not leak oil?

The Motor Company has upgraded the four-inch bore head gasket from the original 16801-07 to one delineated by part number 16801-07A in an attempt to rectify this leakage situation. At the time of writing, the Motor Company was redesigning the gasket once again and there are rumors of a

16801-07C gasket in the works. If true, this means four different gaskets to attack a symptom. The new gaskets have not met with success in alleviating this problem. Of course, how could they? Please see Chapter XV: 2009 Model section, CVO 110 Engine for some Band-Aid H-D responses.

Fixing a symptom does not fix the causal factor.

In Volume I: The Twin Cam, Chapter VIII: The 2007 TC96 Engine, subsection, CVO 110 Rear Head Gasket Leak I ruled out the head gasket surface since I was not aware of the head causing leakage problems in other applications. Well, I take that back as I have now seen and measured .007" warpage on the rear CVO 110 head gasket surface but none on the front head's gasket surface. Hmm, this glaring clue leads, once again to the initial cause of excess heat. Interestingly, in our experience, 116, and 124 cubic inch S&S Cycle engine gaskets

do not fail. S&S uses an oil passage return Viton O-ring part number 50-8034.Viton is a great material to use for engine sealing O-rings, as oil or heat does not compromise it.

Again, there is much speculation and Internet chatter that the rear cylinder steel insert is moving and separating from the cylinder to cause this leak. The reader already knows the truth of this speculation. If it were occurring, there would be more severe problems. Cylinder cutaway pictures found in this chapter destroy the inaccuracy of this and demonstrate what is really going on. However, it is easy to believe there is some degree of separation from the aluminum cylinder as I can hook my fingernail on the cylinder sleeve where it meets the cylinder and the liner.

Under a magnifying glass, it looks as if there has been a disturbance between the liner and the cylinder body. In

addition, impregnating oil seeps out between the two with heat applications. As a general comment on Harley-Davidson entering the high performance world, none of these problems surprises me as every performance parts manufacturer has gone through teething problems to some degree when initially venturing into the big-inch engine field. I do not believe Harley-Davidson will be an exception.

Attempting to rectify this situation with gasketing will not work if the problem(s) prove to be a structural design or manufacturing hitch. A thicker fibrous gasket may alleviate the symptoms. Either way, I am a big fan of an O-ring around the oil passage as it intersects the head and the cylinder. Again, perhaps the CVO 110-cylinder needs a redesign to allow this simple but effective feature.

Hmm, there is another clue. Why do the TC95 and TC103 cylinder re-bores not suffer the same problem as the CVO 110? After all, not all three engine cylinder bores have

enough room to use an O-ring around their oil return head gasket dowels. Yet, only one of the three suffers an oil leak.

Again, the liner top portion is shifting as evidenced by catching my fingernail on the liner ridge where it meets the alloy cylinder body. The H-D three-layer head gaskets need a very smooth surface to function correctly. MLS gaskets, the acronym is for multi-layer system, comprise of three separate layers of stainless steel or aluminum. The Harley-Davidson MLS gasket's two outer layers are embossed in .001" thick Viton rubber-coated stainless steel. Never use any sealer whatsoever with these gaskets. I can see where frustrated mechanics might try sealers as an assist. The MLS gaskets are corrosion resistant, reduce bore distortion, and withstand extreme cylinder pressures. This may well be the reason why MLS gaskets do not blow from the cylinder outwards in the CVO 110 even with the extreme heat present in the rear cylinder.

Unfortunately, the use of Viton rubber O-rings is not possible around the dowel pins, as they will compromise the gaskets sealing effectiveness although cylinder and gasket redesign would handily take care of this irritation. The MLS gaskets need a very smooth roughness average (RA) gasket surface to function effectively.

The earlier Harley-Davidson CVO 110 head gasket uses a Viton O-ring around the oil-return-dowel but the proximity of dowel and cylinder liner lets the O-ring hang onto the shifty cylinder liner.

A Viton O-ring in conjunction with a thick fibrous gasket and fire ring would probably resolve the blowing/weeping situation if the dowel-oil-return hole were centrally located on the alloy gasket surface away from the disruptive cylinder liner. However, it cannot effectively do any sealing because it

hangs onto the sleeve very close to the cylinder bore. This is where the speculation that the head gasket failures are exclusively the fault of cylinder liner movement evolves.

The new MLS gasketing does not use an O-ring and leads our causal diagnosis elsewhere. The MLS gasket has a ridge for dowel sealing and it overlaps onto the cylinder liner.

S&S Cycle, Inc. the stroker and big bore king of the Harley-Davidson aftermarket, uses what appears to be an aluminum center covered with a Kevlar fiber gasket with a fire ring on its inner diameter on its 4.000” bore gasket with never a problem of leakage or blowing that I am aware of. The H-D 3-piece gasket is smooth on both sides with a stainless steel center not bonded to the outer layers like the S&S gasket. The S&S 4.000” bore head gasket has a fire ring that can compress and fiber material to mold and seal more imperfections than

the H-D multi-layer-system (MLS), gaskets can.

At Heavy Duty Cycles, we have been substituting S&S 4.000” head gaskets for all of the H-D gasket series to date.

Harley is hot on the case to correct this problem so new developments can occur at any time.

If the reader looks at a TC88 head gasket, it is thick and fibrous to absorb variances, minor warpage, and a rougher gasket surface. It also has a fire ring to seal compression in, not that compression leaks have been a problem with and of the Twin Cams including the CVO 110. The TC88 gasket also incorporates an O-ring (11273), around the two locating dowel pins on each head to add a further seal. The O-ring provides an excellent seal around the one dowel per head that does double duty as connecting and directing the oil return passage from head to mating cylinder. I mean, what is to

understand here?

Of course, there is sufficient room for the O-ring and the cylinder liner sleeve does not move or rise above the rest of the gasket surface.

A cylinder head redesign would be necessary to incorporate the last two features on the CVO 110 as the cylinder liner bore is too close to the dowel. A thicker gasket may result in a Band-Aid for the symptom but would not thwart the cause of gasket oil leaks.

In my opinion, the new MLS gasket is just too high tech to correct other symptoms such as head warpage although it does seal the shifty top part of the cylinder liner.

The cause, of course, is excess heat in the rear cylinder.

Lower the heat and many of the symptoms would disappear.

On Chris's CVO 110, the head gasket surface was not smooth and had crude machining marks. Tightening down the gasket to specifications may even be pushing cylinder fins away from top liner on the rear head.

Again, here is the kicker; MLS gaskets require smooth, flat, and true head and cylinder deck surfaces to seal. MLS gaskets require a surface finish of 50-micro-inch roughness average (50 RA), or finer for a proper gasket seal. Anything rougher will conflict with the gasket design. For aluminum heads, the numbers are even lower. The typical recommendation today for aluminum is a surface finish of 30 to 60 RA, with the preferred range being from 50 to 60 RA.

For a comparative understanding, honing with #220 grit

stones typically leaves a finish of just over 50 RA. Using #240 grit stones will usually leave a finish in the 38 to 45 RA range; #280 grit stones will generally produce a finish of 18 to 25 RA and #400 grit stones, a finish of 10 to 15 RA. Honing with #600 grit stones can push the numbers down to five RA. A pane of window glass, by comparison, measures about 3 to 4 RA. I mention this because it is important in solving the causal problem.

Furthermore, it looks like the dealer mechanic tried to smooth some high spots on Chris's rear head with a hand-held power grinder disc leaving an RA much too high for the MLS gasket to seal.

In Chris's second dealer rebuild, it is a coin toss to figure out whether the disc grinding leaving an approximate 70-80 RA gasket surface finish or extreme heat blew the gasket first. Both conditions will do it. It is a question of which, came first, the chicken or the egg.

As I have said the thicker, fibrous TC88 style gasket may absorb low spots, high spots or gasket surface roughness but an MLS gasket will not tolerate this.

Smoothness has become a major issue because the difference in thermal expansion rates between an aluminum head and cast iron cylinder liner creates an amount of sideways shearing force and scrubbing action on the head gasket. If the surface finish is, too rough (more than about 60 RA), the metal will bite into the gasket and pull it sideways as the head expands and contracts. The cumulative effect over time can cause a de-laminating effect in the gasket, literally tearing it apart causing it to leak and fail.

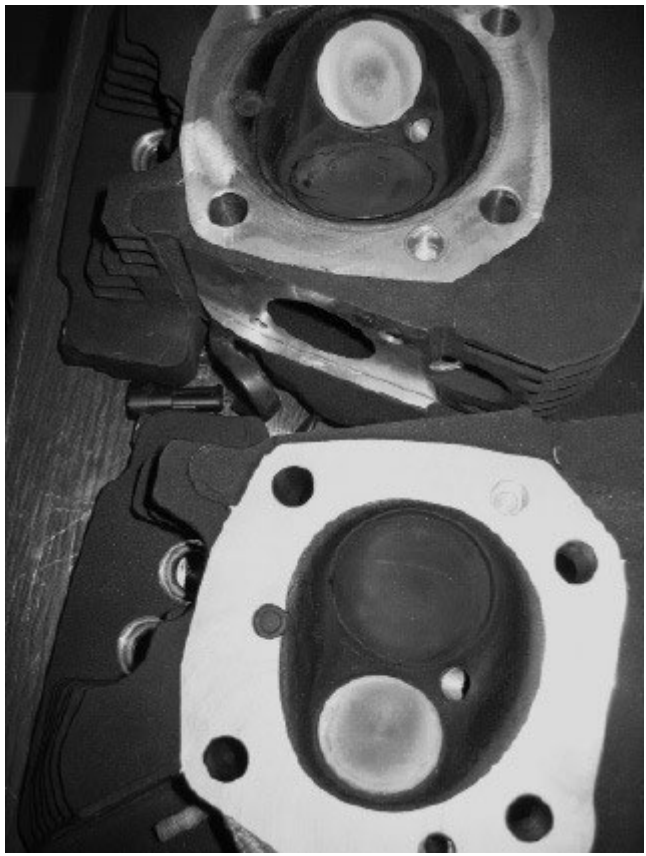
Some aftermarket gaskets can handle a rougher gasket surface finish on these types of engines, but it all depends on

the design of the gasket.

Even so, smoother is definitely better on these engines. The MLS design prevents the gasket from losing torque over time. The composition also reduces the amount of torque that has required on the head bolts to seal the gasket, which in turn reduces cylinder bore distortion for better combustion sealing and reduces internal combustion chamber-ring blowby.

I cannot fault the Motor Company for trying to cure this heating mess with progressively better and superlative gasketing. Even though the gaskets are failing, they are leading edge technology.

CVO 110 Head Gasket Surface



Pictured on top is the front head. There is no warpage on the front head-gasket-surface. Notice Harley-Davidson manufacturing swirl marks from machining on the head gasket surface. We call this process planing. It is a mechanical term describing the process of making a surface flat. The bottom head is the rear one. The surface was warped from inside to out, higher on the inside and lower on the outside gasket surface for a total of .007” warpage. The manual says to replace the head if warpage exceeds .006”. At Heavy Duty Cycles, we planed this surface finish flat and smooth. Not everyone has deep pockets to replace a head because of warpage. The head is good as new when planed flat. Removing .007” from the gasket surface is not enough to affect compression although it will increase marginally. We will plane the front head also to keep compression equal in both cylinders. The round pockmark directly opposite the 12mm spark plug hole is the decompression release hole with the bottom flat showing of the installed automatic compression release (ACR). It is clearly visible on the lower

head because the planning procedure has also removed carbon buildup on the squish The upper head, with careful examination, shows the ACR and decompression hole centered in one side of the squish band. The lower head delineates the shape of the compression chamber perfectly. If we visualize a perfect circle encasing the bathtubbed combustion chamber, the flat area would be the squish band. This single feature helped the Factory meet emissions controls right from the advent of the Evo D-shaped combustion chamber and squish band. The bathtub squish band is a further and superior refinement. The flat-topped piston rises in the cylinder bore to meet the squish band. The air-fuel mixture turbulently forces into the smaller combustion chamber. The turbulence mixes the air and gas so that a higher percentage of burn will occur. Ergo, fewer emissions exit the exhaust because they burn creating energy; improved gas mileage occurs because more of the fuel that is already there burns and this is the primary reason Evos and Twin Cams are so much faster than the inefficient hemi-head Shovelheads. The exhaust valves are

always smaller than the intakes on modern Big Twins beginning with the Shovelhead in 1966. However, the CVO 110 exhaust valve measures at 1.165”. This compares to a stock TC88 and TC96 stock valve measurement of 1.570”. Notice the white carbon texture of the exhaust valves outer heads baked white from extreme heat. The larger intake valve outer heads, 2.075 “as opposed to a stock TC 88 and TC96 1.800” diameters, are dark indicating that the heat is not injurious on that side of the combustion chamber. This may seem strange to the uninitiated but temperature is much higher on the exhaust side of the combustion chamber because of exiting super-hot gases. The intake is cooler because air and fuel entering the combustion chamber is at ambient outside temperatures. The temperature change between the two valves can be well over a hundred degrees Fahrenheit. The oil return hole is close to the exhaust valve. Part of the oil return hole with dowel pin removed is visible at the bottom of the picture in the rear head. The dowel hole in the top head is the locator pin and not the return oil hole,

which is not visible as it is on the opposite side. A partial solution to the oil leakage but not head warpage would be to relocate the oil return juncture away from the hotter exhaust valve and close to the cooler intake valve.

## The Causes

During engine warm up, the H-D electronic fuel injection runs in open loop function (mode), at a 12.0:1 air fuel ratio. The default idle AFR is 12.0:1 for new engines no matter what the engine temperature is even if disconnecting the exhaust O2 sensors.

Once the ECU determines when the engine has heated sufficiently, the EFI goes into closed loop mode at hot idle. The engine will hot idle at a very lean Stoichiometric 14.7:1.

Normal idle speed is 1000 + or – 50 rpm. Cold idle will be

the higher figure or more.

In high temperature mode, the engine drops to 950 rpm and the ECU goes to open loop mode, richening the fuel mixture to 12:1 in an attempt to cool the engine.

Ride Your Bike with a Cold Engine!

Once the rider detects the rpm change, it is time to ride so the air can do its cooling. I never let my bikes sit and idle. I get on them and ride conservatively until warmed up letting the air work its cooling magic. There is no airflow when stopped at idle.

If the engine idles up to 1200 rpm, usually by the rider feathering or blipping the throttle, the ECU retreats back to closed loop mode at 14.7:1, which is too lean for an air-cooled engine especially at idle.

If Idling, Do Not Touch the Throttle!

Leave an EFI throttle untouched by human hand, as the smallest amount of throttle will lean the engine out as long as the ECM is operating in closed loop high heat mode. EFI functions quite well untouched at idle. The interfering human touch only makes a dire situation attempting to cope as best as possible within its EPA limitations much worse.

If the bike accelerates past 1200 rpm at idle, the engine will stumble under high heat duress. The continuation of the lean 14.7:1 AFR condition can drive engine temperature even higher especially in the air-shrouded rear cylinder.

On super hot days or in stop-and-go traffic or a combination of both, the ECM will sense a potential

overheating situation in progress and put the engine into parade duty mode. The ECM will direct the EFI out of closed loop function thus lowering the AFR to an approximate 12.0:1 AFR rich condition. This richer AFR attempts to cool an air-cooled engine that is not getting relief from air circulation.

Cylinder head temperatures can go nuts in this type of situation, much higher sometimes of the acceptable F and my preferable to F. Of course, lots of fiberglass, especially the lowers on the FLHT models only exacerbates a bad situation.

Stoichiometric AFR and Fiberglass do not Mix!

These high temperatures result in two destructive trends; the first is engine detonation, known as the great engine destroyer and the oil, particularly the more vulnerable to heat fossil oil starts breaking down via heat oxidation. A third

problem may also raise its ugly head with idle dropping to 950 rpm, which is a lessening of pumped oil supply, particularly to the cylinder walls and mating piston thrust skirts. This may result in the piston oil-cooling jets squirting only around 90% of oil. Of course, this situation compounds the others since less oil delivery, once again equates to higher cylinder temperatures. Higher cylinder temperatures may heat-scuff the piston thrust surfaces, fore and aft.

If the reader thinks exhaust bluing was bad in the past that was in the good old cooler days of richer AFR ratios where closed loop EFI was only an esoteric term few understood. Therefore, the 2007 and later engines as well as the 2006 Dyna guinea pig introduction model have very hot exhaust systems that must cope with the world of closed loop Stoichiometric AFR ratios.

An AFR of 14.7:1 and higher, particularly on the rear cylinder, produce yet another of a litany of unwanted

destructive, unproductive or annoying symptoms while in closed loop cruise mode, which is poor throttle response while cruising. In cruise EFI closed loop mode, the narrow band O2 sensors send exhaust content electrical signals to the Delphi ECM. They signal the ECM to deliver an air fuel ratio of 14.7:1 through the EFI.

This is what the EPA demands and makes Harley-Davidson compliant but creates all the heat prostration that leads to the head gasket oil-leak breakdown. The rear cylinder temperature heat, a result of leanness and exacerbated by shrouding from the front cylinder contributes to the heat because of lack of airflow combining with hotter air coming back from cooling the front cylinder. Preignition and resultant detonation contributes much to the rear cylinder problem because of excess heat over front cylinder. In fact, the Stoichiometric fuel ratio of 14.7:1 goes up to 14.8 to 16.0:1 on the rear cylinder due to insufficient cooling from airflow

leading to the detonation.

The intense heat starts to expand the .625 inches of cylinder liner not tied in by spiny lock construction. The rear head-gasket surface will begin to warp in a plane between the head intake and exhaust ports. A 3-piece MLS gasket that depends so much on a smooth gasket surface cannot protect the rear oil-return-dowel, in line with this warpage plane.

In addition, Harley-Davidson keeps inventing or using new leading edge gasketry to treat the symptom not the cause as per usual with the CVO 110.

## The Band-Aids

My intention is not to demean the following Harley-Davidson and aftermarket assists in dealing with the high heat situations that we now face because of closed loop mode EFI

seeking Stoichiometric 14.7:1 air fuel ratios on an antique engine design. They are all necessary and welcome additions to the fight against extreme heat. However, they are not cures in themselves because the cause of the extreme heat fails addressing.

I have already discussed parade duty open loop EFI mode to combat hot idle conditions. However, it bears repeating that the engine needs air-cooling; so idling is not a good option with the new EPA demands. Ride your bike immediately and conservatively in the wind until warmed up.

The Engine Management System

Harley-Davidson has made huge strides in this area, both to modernize their engines and to meet increasing EPA demands. The progressively changing and improving Delphi

EFI is light years ahead of the inefficient and slow Magneti-Marelli EFI systems used on the first Twin Cams. Fuel efficiency is at an all time high with emissions at an all time low.

The ESPFI (electronic sequential port fuel injection) has followed the increased sophistication incorporated in the experimental 2006 Dynas that works extremely well given its mandate and conditions of operation. The 2006 Dyna introduces the use of a new feature, an O2 oxygen sensor that makes closed loop mode EFI a reality. This oxygen feedback sensor locates out of visual way behind each header exhaust pipe.

The sensor measures exhaust gas composition and sends signals back to the electronic control module that in turn signals to the electronic fuel injection to adjust fuel delivery for optimal air/gas mixture. This, of course, is within EPA emissions guidelines especially in cruising closed loop mode

with the Stoichiometric air-fuel-ratio of 14.7:1.

The bandwidth of the H-D O2 oxygen sensor is narrow but allows changes within many naturally aspirated performance parameters.

Naturally aspirated refers to carburetion and EFI where air breathing operates within the natural laws of physics like vacuum (negative pressure). However, an increased bandwidth 02 sensor like that used with some aftermarket self-tuning ignitions will make this ESPFI cook in conjunction with supercharging and turbocharging. Boost from forced aspiration also known as ram flow can change cylinder pressures and rpm capabilities rapidly.

Wide bandwidth 02 sensors increase reaction time. They can deal with and relay these fast changes to prevent possible

injurious extreme heat situations.

The previous angled fuel injectors are now at more proper 25-degree angles.

The 2006 Dyna TC88 introduced a Delphi 46mm EFI venturi throttle body. The 2007 TC96 models carried over the use of 46mm bodies with a 50mm Screamin' Eagle option. Then the 2007 CVO 110 also fits with the 46mm throttle body…duhhh! The Screamin' Eagle upgrade is now a 58mm body, which in my opinion is making a mistake going the other way to too big from too small for the CVO 110. The 58mm EFI throttle body needs at least 100 rear wheel horsepower to come into its own. Sad to say, I don`t believe the CVO 110 fits into the rear wheel 100+ horsepower club. This 58mm body will not go to waste, as it will be ideal on a larger engine.

Of course, this all pales in comparison with the anemic and paltry 38mm throttle body used on the first Magneti-Marelli electronic fuel injection systems used on Harley-Davidson 80ci Evolutions and the first TC88's.

H-D Assists

The EITMS (engine idle temperature management system), also known, as idle temperature cool down turns off the rear cylinder's fuel injector at hot idle. Why would we do this when gas is the cooling agent and air causes the villainous heat? Burning gas assisted by maximum air generates the heat from too high air fuel ratios. In this case, the Factory uses air as a cooling agent because there is no ignition of gas in the rear cylinder during

The 2007 to mid-2008 engine idle temperature management system in simple terms offers a cooling feature

for hot idle conditions. This engine calibration procedure is available for all FLHT touring models plus police specific FLHTP, FLHP, and FLHPE motorcycles. It is also available for Dyna and Softail models.

This option is available for all these models free of charge from your H-D dealer. EITMS meets the needs for those that ride in very congested traffic on a regular basis or need to leave their bikes idling as in a police situation. It also assists overheating situations, when the bike returns to idle. The calibration offers limited rear cylinder cooling when cylinder temperatures reach a certain level.

Once EITMS activates, the rear cylinder still functions without fuel. Air still sucks in on the intake stroke, compressed on the compression stroke, decompressed on the power stroke, and exhaled on the exhaust stroke. The air acts

to cool the engine rear cylinder while only running on the front cylinder. The EITMS will only function when there is high engine temperature, the engine is at idle speed, the bike is not moving, and the transmission is in neutral or if in gear with the clutch lever pulled in thereby disengaging the clutch. When all of the above conditions meet, the rear cylinder will stop firing. EITMS will allow supply of fuel to the rear cylinder if temperature decreases, the throttle is increased, the bike moves forward, the clutch engages or the bike shifts out of neutral with the clutch engaged then the rear cylinder and thus the engine will begin running normally.

The potato-potato idle sound will be different since only one cylinder is contributing one potato to the noise rhythm while there is the sound of the whooshing of air from the other. This will also affect the exhaust smell. I mention the above to allay fears that not all is right if not prepared for these EITMS normal symptoms.

This free one-time activation and one-time deactivation if the rider finds it unnecessary or simply dislikes the feature is available for the models and years listed above. The 2008 Screamin' Eagle models have the calibration activated from the Factory. The owner can have it deactivated by request at no cost. Quote Service Bulletin M-1191; labor code 8202; customer concern code 9203 and condition code 9110 although referring to M-1191 will do the trick.

In 2009, the FLHT Engine Idle Temperature Management System becomes rider controlled. The EITMS feature located in the stock ECM or electronic control module can turn on-or-off via the throttle grip. Once enabled, the rider can turn off fuel delivery to the rear cylinder when the bike is The rear cylinder then becomes an air pump as cooling air introduces and expels with the four engine strokes. Once the throttle turns for acceleration away from idle, the EITMS turns off automatically. The 2008½ and 2009 EITMS calibration cannot

be used on early-2008 and previous years touring models.

There is new ECM fuel and ignition calibrations for 2009.

The Factory has also developed a fan (part # 91550-00C, chrome; 91531-00D, black), that installs on touring FLHT's to lower engine oil tank temperatures a claimed 20 degrees Fahrenheit measured at the oil tank when using the fan. 4

Harley includes its Premium 8-core oil cooler on the CVO 110 to assist in combating higher oil temperatures.

I believe that H-D only has a 9.3:1 compression ratio on the 2007 and 2008 CVO 110 to combat higher cylinder temperatures by reducing compression. This is necessary for combating EPA current and future standards but is a step in the wrong direction for a performance engine. Remarkably,

compression reduces on 2009 CVO 110 models to 9.15:1 and 9.2:1 on the Ultra Glide model.

Further Harley-Davidson assists need to be in the redesign department so engine components will not wither in the onslaught of very high AFR fuel ratios.

Aftermarket Assists

BAKER Drivetrain (www.bakerdrivetrain.com) now has the +1 Oil Pan that fits 1993-2008 FL touring models increasing oil capacity a full quart for a reduction in operating temperatures and an increase in engine durability.

Everything is about temperature because it does not require a rocket scientist to understand that the cure is about less heat. Any method or more correctly methods offering an assist to a cooler operating cylinder is welcome. BAKER claims

a F reduction at F.

The 8-core turbulated H-D oil cooler is a great idea for any Twin Cam. It is a stock and very necessary component on the CVO 110. As I said above, its addition to the CVO 110 is an attempt to control cylinder head temperatures in addition to the anemic compression ratios that range between 9.15:1 and 9.3:1. Turbulation mixes the oil for superior heat dissipation. Coolers without turbulation only cool the outside laminar flow of oil, an inefficiency that leaves central lamination as hot leaving the cooler as when entering it. However, I feel that only one oil-cooler is not enough. As Bob Wood has taught me, we need more cooling to lower cylinder head temperatures to between to 5 to control engine detonation, even though literature list 225-degrees as a manageable temperature.

It is not a good idea to run two H-D oil coolers in tandem because of pressure drop. Oil restriction causes pressure drop. We do not want to risk starving the engine for any oil whatsoever. I discuss the performance necessary oil coolers, thermostats, turbulation, pressure drop and other factors in Chapter VI: Oil Coolers. Externally, I recommend running two 10-core Jagg oil coolers in tandem on any performanced Twin Cam. I have them on my personal bike and the combination works very well.

Internally, we need to lower the air relative to gas or raise the gas relative to air.

I have discussed bigger injectors, as more gas will lower injurious rear cylinder temperatures. A bigger injector like that used in the 50mm throttle body for the rear cylinder will keep the same time-on pulse width but with more volume of gas squirted, so more gas can effectively lower AFR.

On Chris' CVO, I will be installing the complete 50mm throttle body with both the larger injectors.

Colder spark plugs are another small assist to the equation. Spark plugs come in a variety of heat ranges. Running a colder plug in the rear cylinder is desirable in the case of the CVO 110. Any current or yesteryear Shovel owners are well acquainted with this procedure as wrenches have been using different heat range spark plugs for a long time.

Changing the exhaust can also remove more heat from the engine or conversely help contain it. We are into rocket science with exhaust technology and it is not in the realm of mechanics or Factory Technicians to design or modify specific purpose exhausts. I will be discussing exhausts at great length in Chapter VIII: Eliminating the Performance subsection,

Exhaust Systems.

## Redesign the CVO 110-cylinders

The CVO 110-cylinders need redesign to cope effectively with the heat. In my opinion, Harley-Davidson cannot use the same techniques on the CVO 110 as used on the TC96. The cylinder bore is just too big relative to the placement of the cylinder studs and oil return dowel.

I know, I have said there are no overt symptomatic problems associated with the front CVO 110 head and cylinder; I feel they are lurking just under the surface so to speak. The safety margin is thin. In other words, adverse conditions may push the front cylinder over the brink to join the rear one with its myriad problems.

I also feel that these cylinders, front, and rear would be

able to cope without redesign if air fuel ratios could lower in closed loop mode.

The Motor Company is between a rock and a hard place. They need to use newer technology to cope with Stoichiometric induced heat especially when shrouded from the air stream. They cannot simply lower air fuel ratios like the aftermarket may do or they will not be in EPA compliance.

In any case, the component parts that must cope with Stoichiometric air fuel ratios that escalate well above 14.7:1 AFR due to shrouded heat inducing ratios approaching 16.0:1 need redesign. This includes the head alloy composition so it cannot warp under these conditions.

This situation is a harbinger of the future. In a non-

traditionalist setting where functionality rules, a water jacket and cylinder configuration redesign would easily win out as the solution to this excess heat. Currently this is not an option but it is necessarily coming…sooner rather than later.

Relocate Oil Return Juncture

If observing the CVO 110 bathtubbed combustion chambers, the rear heads oil return passage locates next to the exhaust valve on the head gasket surface. The exhaust valves are always smaller than the intakes on modern Harley-Davidson's. The carbon texture of the exhaust valves outer heads bake white from extreme heat within the CVO 110 combustion chambers.

The larger intake valve outer heads remain a dark and familiar carbon color indicating that the heat is not injurious on that side of the combustion chamber. This may seem

strange to the uninitiated but temperature is much higher on the exhaust side of the combustion chamber because of exiting super-hot gases.

The intake is cooler because air and fuel entering the combustion chamber is at ambient outside temperatures. The temperature change between the two valves can be well over a hundred degrees Fahrenheit.

A partial solution to the oil leakage but not head warpage would be to relocate the oil-return-juncture away from the hotter exhaust valve and close to the cooler intake valve. This involves an easy but more complex machining procedure but is doable.

Move the Oil Return Dowel away from the Cylinder Liner

Lowering heat, which is the main culprit, is a difficult undertaking for those producing an air-cooled engine with in-line front to rear cylinders. The shrouded rear cylinder bathes in the heat dissipation of the cylinder in front.

Remanufacture cylinders with more meat to accommodate the 11273 O-ring or similarly a more effective gasketing technique that can absorb some warpage of parts.

Use Cylinder Liner Spiny Lock Construction to the Top Deck

As I have written, the spiny lock construction in the CVO 110-cylinders only go to within .625” of the top cylinder deck allowing expansion and contraction to disrupt the head gasketing surface that must be perfectly flat. Change this specification to a minimum of .094” (⅜”), from the cylinder deck top as is found with the TC88 cylinder construction.

This will prevent the cylinder liner separation found on the rear cylinders of the CVO 110 because of too-high AFR ratios.

## Equalize and Improve Cylinder Stud Tie Down

The TC88, TC95, TC96, TC103, and the TC110 all use the same cylinder stud pattern, with the same distance between each.

Big inch engines that must cope with closed loop Stoichiometric AFR should go to an equidistant 5-stud system to replace the 4-stud cylinders in use since 1984. S&S Cycle Inc. has done this with its innovative X-Wedge engine that meets emissions standards up to and including the 2010 more stringent dictates.

Finally, moving the cylinder studs out further into the

middle of the gasket surface will assist much in the even and consistent application of gasket pressure. This will equalize torque pressure specifications. It will add the extra support needed to cope with higher temperatures.

The 2007-2009 CVO 110's still use the Twin Cam 88 cylinder studs, part number 16834-99A in use since 2003 when they replaced the 1999-2002 cylinder studs, part number 16834-99. I will replace the stock studs with higher tensile ones like those provided by S&S, part number 31-2340.

Ensure Cylinders are a Plug Fit into their Spigots

As I have said, there is an approximate .028" gap between cylinder spigot hole and the cylinder flange, which is the extension of the cylinder liner that protrudes from the base of the cylinder into the spigot hole. This extension is necessary

to both provide support for the piston skirt fore and aft thrust surfaces, particularly as the piston changes axis at bottom dead center. The piston must suddenly stop and change direction guided by the cylinder flange. The outer side of the flange, if a plug fit in the engine casing cylinder spigot hole will also provide some rigidity and support to the whole moving mechanical process.

The aluminum engine casing expands more and faster than the cast iron cylinder liner so a tight plug fit will not cause a conflict with expansion and contraction of mating parts. It is an easy manufacturing procedure to tighten specifications to a plug fit into engine casings.

Duplicate the TC88 Cylinder Design into the CVO 110-cylinders.

Make the cylinder CVO cylinder liner thicker if it makes

people feel better. Extra rigidity has always assisted and never hurt and engine`s functionality.

Use the same cast iron alloy in the CVO 110 sleeve as with the TC88.

Restore Greater Finning Area

The CVO 110-cylinder finning area is less than that of the TC88. As a minimum requirement, restore the finning area to that of the TC88. Preferably, the redesign would extend area finning even more. This includes, matching the cylinder head finning.

Oil Cooling

The Premium Harley-Davidson 8-core turbulated oil cooler is one of the best on the market. However, it is not enough in my opinion. In this case, more is better as long as oil restriction through the cooler is balanced. I feel a 14-core oil cooler or dual 10-core oil coolers with less pressure drop will be a necessary assist in combating heat.

Increase Oil Reservoir Capacity

BAKER Drivetrain leads the way on this one with its 1 + oil reservoir. I feel Harley-Davidson should duplicate this feature as a standard item on all models that must meet Stoichiometric AFR.

This excludes the V-Rod and the new 1125R Buell because they already cope wonderfully with excess heat due to their water-jacketing and cylinder angle reconfigurations. Sorry to be the bearer of bad news for the traditionalists like me, but

these bikes is the future of Harley-Davidson.

## Synthetic Oil

Fossil oil should become an oil of the past for air-cooled engines coping with closed loop Stoichiometric AFR.

The superior lubricity of synthetic oil should become the mainstay of these engines. Synthetic oil withstands much higher temperatures than fossil oil before beginning to break down because of heat-induced oxidation.

## Rear Cylinder Fans

Unless there is radical redesign of the engines, I feel a rear cylinder fan that is now an option should become standard equipment. The Harley-Davidson rear cylinder head fan's part numbers are 91550-00C for chrome and 91531-00D for black.

## Cylinder Head Machining

I have already mentioned using an alloy more resistant to extreme heat than is currently utilized to resist warpage.

The machining of the head gasket surface needs addressing, in my opinion, from quality control especially if MLS gaskets that require a roughness average of about 50RA are used.

## Solving the Tendonitis of Lean Burning Air

As I have said before, H-D has a real conundrum here. It cannot meet EPA mandates and lower closed loop AFR at the same time. The Factory can only resolve this issue by adapting

existing water-cooling and other similar distasteful technologies and destroy the traditionalist love most have for its now beloved dinosaur design.

Preferably, the Motor Company can do the redesigns mentioned above in addition to exploring new technologies that will ably cope with the new heating issues that are not going away.

For us, the consumers, our options are more flexible. Our solution is to lower temperatures by lowering the air fuel ratios below 14.7:1 during hot idle and cruising situations.

The SERT (Screamin' Eagle Race Tuner can lower AFR to approximately 14.2:1 which is adequate but Power Commander can take the AFR down even lower if necessary.

Both the SERT and the PC require skilled technicians to tune them.

EFI Terminal Velocity and the carbureted IST ignitions are self-tuning. After installation, they do all the tuning work and constantly change tuning according to information received from the oxygen sensor(s).

There is some controversy over the stock narrow band Oxygen sensors. I too have fallen victim to this but no more. Narrow band oxygen sensors do not need replacement with wideband oxygen sensors except in specific applications where response time is critical. Therefore, the stock narrow band sensors are adequate for most performance work.

All of the above ignitions and many more in the marketplace adapt to performance changes. They are not a onetime install for specific changes like the Screamin' Eagle

burns also known as downloads that become obsolete as soon as performance parts are changed.

There are two main types of fuel-ignition modifiers with many varieties within each.

Input modifiers locate between the oxygen sensor(s) and the ECM. Terminal Velocity and SERT are examples, where modifying information from the sensors to the ECM changes the air fuel ratios dictated by the ECM to the EFI.

The fuel table, matrix, or maps, take your pick as these terms all mean the same thing directly relates to the ignition table, matrix or map. Therefore, both fuel and spark modifies to the EFI and the ignition to coordinate spark and fuel.

Output fuel modifiers intersect the ECM and the EFI.

Oxygen sensor information goes unchanged into the ECM. The fuel directives from the ECM to the EFI modify by increasing the pulse width of the EFI injectors. Pulse width is how long the injectors stay open. Therefore the air fuel ratio lowers (richens), because the output modifier lengthens the injectors pulse width bands. Output modifiers can be limiting because all except the Power Commander can only change fuel but not the ignition table. This is not necessarily a put down since fuel modification is the primary change we are after in this particular situation. Power Commander is unique in that it has an ignition component.

The result of fuel modification will be a head gasket that retains its integrity and is able to do its job.

The AFR going above Stoichiometric 14.7:1 is the true villain. We can lower temperatures with assists such as EITMS, cylinder cooling fans, synthetic oil, larger capacity oil tanks, and dual 10-core oil coolers.

However, the cure will revolve around one or a combination of two things; utilization of components that can withstand and function within higher operating temperatures and the second is to lower temperatures. If we can keep the rear cylinder at Stoichiometric AFR, we can probably live with this and obey closed loop emissions dictates.

Lowering below Stoichiometric will give increased engine longevity along with performance improvements.

Another way to lower cylinder head temperatures is to further lower compression.

Do Not Lower Compression

An obvious way through part of this debacle would be to lower the compression in the CVO 110 as this would lower cylinder pressure.

In physics, pressure and heat are synonymous. Expanding heat from burning fuel drives the piston downward on its power stroke. Increasing pressure during the compression stroke via an upward moving piston creates heat. If this heat grows too severe, preignition and the devastating detonation are destructive events about to happen.

So, why should the Factory not lower compression? The raison d'être is that the whole purpose of creating a 110 cubic inch Harley-Davidson is power. We do not want to decrease power by lowering compression, now do we?

Increase Cam Duration

The stock CVO 110 .550” lift cams have durations of on the intake and the exhaust

The S&S .585G gear drive cams that I use in Chris’s CVO 110 have a higher lift of .585” that I increased further to .603” lift via a longer roller rocker arm ratio. The duration figures for these cams are on the intake and on the exhaust. This is not a radical departure as the stock CVO cams have some emissions controls built-in by decreasing cam duration.

Increasing duration allows for higher compression. This is because compression does not start to build until later in the compression stroke because the valves are open longer at the beginning of the upwards stroke. I will explore this subject much more in Chapter VII: Torque and subsection, Compression and in Volume VI: How it all

Compression will also be a subject in succeeding volumes on the Shovelhead, Sportster, and the Evolution, particularly as it applies to these differing levels of technologies and cylinder head designs.

## Porting and Polishing the CVO 110

The heads, front OEM 17250-07, and rear OEM 17248-07 have big expanding intake ports similar to a Shovelhead to accommodate bigger intake valves.

The valve seats are thinner in depth and width as the following chart will demonstrate.

### Comparing CVO 110 Valve Seat Dimensions

The Screaming Eagle heads on the CVO 110 only have a 3-angle valve seat cut. Optimally we want a 7-angle cut but many times, we cannot do this because the seats have already been radiused out at the Factory. However, a 5-angle cut to each valve seat face will further round the entranceway for the incoming fuel charge and initial departure port for the exiting exhaust gases. Please see Chapter VIII: Eliminating the Performance Obstruction for in depth information on porting and polishing.

## Performance Bonus of a Cooler Air Fuel Ratio

This one is short and sweet. If we lower the AFR to about 14.2 or thereabouts depending on circumstance, not only will the blowing head gasket syndrome disappear but the bike will be very much faster too.

Comparing 110c.i Cylinders

Pictured is H-D CVO 110 cylinders and pistons on the left and Axtell 110 cylinders and pistons on the right. The Axtell cylinders look larger and in some respects are. They exude strength, rigidity, and stability. CVO 110-cylinder liner flange thickness is approximately .100” compared with Axtell thickness of .120”. CVO 110 and Axtell 110 internal top liner thickness is .150” and the thickness of the CVO 110 internal bottom liner is .160” while Axtell measures in at a beefy .200”. Most strikingly, the cylinder bottom flange-length of the CVO 110 is a measly 1.000” comparing to the very beefy Axtell length of 1.630”. The much longer flange length provides more rigidity, support, and stability for both the cylinder and its piston. The piston at BDC must stop, change axis and direction. The piston needs all the support it can get in this area with all the physical forces with which it must contend. The Axtell cylinders have two less cooling fins but they are much thicker and beefier than the Harley CVO 110 cylinder. The smallest fins at the bottom of the Axtell cylinders are missing but otherwise have the same area as the

CVO 110. The thicker Axtell fins have supports evident in both pictures but more pronounced in the lower one between the fins on the sides to add extra rigidity and support. This one tough cylinder is all business. The longer flange must be relieved in two places to allow parts movement and functioning. The top picture shows a cutaway in the right side of the flange to accommodate the piston cooling jets that occupy space in the right side engine case cylinder spigot holes. The lower picture shows a milder cutaway in the center of the cylinder to allow connecting rod clearance. The connecting rods rotating at bottom dead center rotationally change axis and direction. Their combined angularity in relation to both the flywheel assembly and the cylinders is greatest at this point. The longer the stroke, the more exaggerated this female and male connecting rod overlap angularity will be. Anyone familiar with stroker building will recognize this feature.

CVO 110 Axtell Aftermarket Cylinders

Another cure that I am exploring with Chris's CVO 110 is to get rid of the stock problematic cylinders and replace them with solid aftermarket ones.

I have chosen a cylinder set with pistons, part number 200-215S, from Axtell Sales of Des Moines Iowa, also known as Axtell Dyno Master Performance Product's (www.axtellsales.com).

There is one wrinkle for this particular application. The Axtell cylinders and reverse dome pistons are for use with an 85cc combustion chamber. The CVO 110-combustion chamber is 95cc. Axtell is on top of their game and substituted flattop pistons leaving Chris with a 9.65:1 compression ratio. It is a pleasure talking to people where the information and technical knowledge is second nature.

As the guys at Axtell say, “A thicker, more stable sleeve than any other cylinder on the market…This 4” bore cylinder was designed to work in harmony with the 4 ⅜” stroker motors that have become very popular… This cylinder and piston kit has been known to help quiet engines and maintain excellent oil control…all stock components will work because this engine retains stock engine height. The 110c.i kit comes complete with Axtell Cylinders, forged pistons, moly-rings and MLS gaskets. Reverse dome piston yields 9.7:1 compression ratio.” 6

So, I am happier now that I have also upped the compression a little from the anemic 9.3:1 to 9.65:1 plus a little more because of planing the head gasket surfaces flat with a .007” cut although the .007” planing will have little effect.

Axtell 110 Cylinders

When comparing H-D CVO 110 cylinders and Axtell 110 cylinders, the Axtell cylinders look larger and in some respects are. They exude an aura of strength, rigidity, and stability.

These cylinders are confidence building for the mechanic looking for superior products to combat the ravages of excess heat.

The CVO 110-cylinder liner flange thickness is approximately .100” compared with Axtell thickness of .120” while both CVO 110 and Axtell 110 internal top liner thickness is .150”. The thickness of the CVO 110 internal bottom sleeve liner is .160” while Axtell measures in at .200”.

Conspicuously, the cylinder bottom flange-length of the CVO 110 is a measly 1.000” comparing to the very supportive Axtell length of 1.630”. The much longer flange length

provides more rigidity, support, and stability for both the cylinder and its piston.

At BDC, the piston must stop, change axis and direction. The piston needs all the support it can get in this area with all the super stress with which it must contend.

The Axtell cylinders have two less cooling fins but they are much thicker and beefier than the Harley CVO 110 cylinder. The smallest fins at the bottom of the Axtell cylinders are missing but otherwise have the same area as the CVO 110. The thicker Axtell fins have vertical pronounced and thick webbing supports perpendicular to the horizontal finning on either side of the cylinder to add extra rigidity and sustainability. These strong cylinders are all business.

Axtell Cylinder Flange Relief

The .630” longer Axtell cylinder flange must have relief in two places to allow parts movement and functioning.

In mechanical parlance, the synonymous terms relief or relieve means to cut away interfering metal to allow clearance between functioning parts. On the right side of the flange as it sits in the engine looking forward from rear to front, an area of the flange relieves to accommodate the piston cooling jets that occupy space in the right side engine case cylinder spigot holes. A milder cutaway in the center of the cylinder allows connecting rod clearance.

The connecting rods turning at bottom dead center rotationally change axis and direction. Their combined angularity in relation to both the flywheel assembly and the cylinders is greatest at this point. The longer the stroke, the

more exaggerated this female and male connecting rod overlap angularity will be. Anyone familiar with stroker building will immediately recognize this characteristic.

Machine Interfering Top Case Bolt

If boring the spigots with the top stock case bolt installed, the boring tool will cut into the bolt. This can result in a loss of tensile strength of the bolt, which may allow a possible oil leak. If re-installing this machined case bolt incorrectly, damage may occur to the cylinder flange and mating piston.

The stock Twin Cam casing top bolt, which is the center bolt between the cylinders bolts through the left casing into a threaded portion of the right casing. In other words, the bolt does not go right through both case halves.

Axtell, quite rightly recommends eliminating this system in

favor of the case bolt going through both case halves for superior and more consistent clamping. Therefore, before boring the cylinder case spigots, remove the piston oilers from inside the right case spigots. Drill the right case top partial bolthole through with a ¼" drill bit, using a drill press or milling machine. Furthermore, spot face the outside of the cases around this hole to allow a flat sealing surface.

Now, reassemble the engine casings following the H-D manual specifications except for the top hole. Install the new ¼" bolt and nut with the bolt head on the right side case and the nut on the left one. Torque this top bolt to 85 in-lbs not ft-lbs.

If machining the top case bolt, instead of following the above procedure, ensure the bolt positions correctly through the casing halves so as not to obstruct the cylinder spigots. If

there is incorrect positioning, the bolt will distort the cylinder flange spigots causing grievous harm to piston and cylinder alike. 7

Boring the CVO 110 Engine Case Spigots for the Beefier Axtell Cylinders

The CVO 110-cylinder spigot flange has an outer diameter of 4.180”. Axtell is 4.240” due to the thicker cylinder sleeve flange. For referencing, the TC88 is a much smaller 4.055”. The CVO engine casing spigot holes measure at 4.208” for the sloppy cylinder spigot fit of .028” (4.208” – 4.180” = .028”).

Machining is necessary for all Axtell big bore kits. The Axtell flange outer diameter measures at 4.240” and therefore, will not slide into the smaller CVO 110 engine casing spigot hole of 4.208” (4.208” – 4.240” = minus .022”).

After installing the case bolts holding the engine casings together to specification, place the assembled casings into a milling machine or boring bar. Bore the case spigots to 1.625” deep. Do not exceed this distance, as there is a danger of cutting into existing oil passages.

Our Axtell flanges measured in at 1.630”. The base gasket will take up this discrepancy of .005”. The 4.000” bore casings in the CVO 110, require boring the spigot holes to between 4.210” and 4.215” diameter to 1.625” deep. However, our Axtell cylinders have a larger flange diameter than the prescribed spigot hole size. Our Axtell cylinders have a diameter of 4.240” so we bored out Twin Cam casing spigots to between 4.250” to get a snug plug fit.

If the cylinders were cast iron or ductile we would decrease the spigot hole sizing .005” to 4.245” since the

expansion rate of these cylinders is less than the aluminium casing.

For reference only, Twin Cam kits with 4 ⅛” (4.125”), bore pistons require a 4.310 to 4.315” spigot bore also 1.625” deep. Then step bore the cases from 1.625” deep .250” further at a bore diameter of 4.160” 8

Comparing CVO Gaskets

The two gaskets on the left are Cometic gaskets

(www.cometic.com) for use with the Axtell cylinders I am replacing the stock CVO cylinders with. The three gaskets on the right are top, a Twin Cam cylinder base O-ring. Use of this O-ring has proven successful over the years and combines with two dowel O-rings the same as in the lower head gasket. The middle head gasket on the right is the CVO 110 second-generation multi-layer-system (MLS), gasket that H-D hoped would rectify the oil-leaking situation. This three-in-one gasket has a metal center with a rubber Viton coated metal gasket on either side. The two metal rings each on a gasket appendage opposite to each other on either side of the bore hold the multi-layer gasket together. This gasket will not tolerate a rough or warped gasket surface. Here are the three major problems with the CVO 110-rear head gasket sealing correctly. The head gasket surfaces that I have thus far observed are not always smooth enough to allow an MLS gasket to work properly. Secondly, the CVO 110 rear head has a tendency to warp under the extreme heat of up to 16.0:1 air fuel ratios in the rear cylinder. Thirdly, the shifty CVO 110-

cylinder liner at the top of the cylinders creates a bump that once again, a MLS gasket cannot tolerate effectively. Interestingly, the thick and fibrous TC88 head gasket with a fire ring to seal compression into the cylinder shown bottom right will help answer the three concerns listed above. It can accept a rougher gasket surface not that it should have to do this. The TC88 style gasket can also accept more head warpage without leakage not that it should have to do this. Moreover, the fire ring can cope with a shifty cylinder liner not that it should have to do this. Lastly, the two OEM 11273-cylinder dowel O-rings positions in the TC88 gasket. I place an extra one in the middle of the gasket for better viewing. This will effectively seal oil within the oil return passage. The H-D MLS gasket, middle right looks so identical to the Cometic (554-241), head gasket, middle left that I am going to say that they are. Therefore, I will use the same MLS gasket that fails on the H-D CVO 110 cylinder to help successfully deal with the problem using an Axtell 110 cylinder that has a smoother gasket surface and a cylinder liner that does not

move. There is a similar but different head bolt tightening procedure. Guess what, H-D's next new attempt after the failure of the MLS gasket through no fault of its own is similar to the TC88 gasket! There is no part number for these gaskets at time of writing but markings on them include "IVAN-AC" and "This side up 4.000". The top left base gasket replaces the H-D O-ring on the Axtell 110 cylinder base. This metal-coated Cometic gasket has sealing ridges around the two locating dowels, one of which serves as the oil return passage also. Another sealing ridge is evident on the inner diameter of the base gasket.

## Boring and Honing Cylinders

### Piston Clearance

Measure piston thrust surface skirt .500" up from the bottom.

On a piston with a clearanced skirt such as the one we are using in our CVO 110 rebuild, there is a different procedure since the clearancing interferes with the above measurement process.

1. Measure the un-clearanced piston on the thrust surface directly under the oil ring.

2. Then, measure the clearanced piston in the same place to ensure both pistons are the same size.

3. Now, measure the un-clearanced piston .500” up from the bottom on the thrust surfaces.

4. Use this measurement plus specified piston-to-cylinder

tolerance to determine final cylinder bore size. Thus, the critical area immediately under the oil ring and ring lands will have adequate clearance.

For example, Axtell recommends .002” to .003” clearance with their forged pistons when fitting to aluminum cylinders, which all Evos and Twin Cams use. This clearance is substantially more than in use for the Twin Cam cast hypereutectic pistons. Axtell recommends .0025” to .0035” for the same forged pistons when fitting to cast iron cylinders, which all Knuck, Pan Shovel and Ironhead XL engines use.

JE Pistons (www.jepistons.com) adds in their instructions that specifications are gauged at F. Their piston-to-cylinder bore tolerances also depend on bore size with TC88 and TC96 bores which are 3 ¾” and TC95 and TC103 bores which are 3 ⅞” requiring .0025” to .0035” while a 4.000” bore like the CVO 110 requires .0035” to .0045”.

If introducing forced induction like super or turbo charging or nitrous oxide then further additional clearance of .001” to .003” (is required) depending on application and once again, bore size. 9 The point is forced induction requires more clearancing.

Always consult your H-D manual for tolerance specifications when using their stock OEM pistons. Consult Screamin’ Eagle instruction sheets supplied with their piston kits when using their aftermarket pistons. Finally, when using Axtell, JIMS, S&S or any other excellent manufacturer or supplier of quality pistons, always adhere to their specifications, boring and honing instructions; break in procedures and oil recommendations if any. 10

Piston Rings

Do not overextend rings as this may cause stress points that can lead to eventual breakage inside the engine.

- Molybdenum-faced or simply moly top rings have best use in freshly bored and honed cylinders.

- Cast top rings are best for deglazed cylinders. Deglazing is a honing procedure to remove glaze from cylinder walls and re-crosshatch them.

- Moly rings have a lighter color on the cylinder-face of the ring than the side-faces.

Moly is harder than cast.

Look for markings on the two top single-rings.

- A pip mark or indentation or any other marking on one side-face indicates that ring installation is with that side up towards the top of the piston.

- A beveled inside edge on the top ring compression ring always faces

- If there are no markings or a bevel, the ring may install either way.

- If there is a bevel on the bottom compression ring, which is the middle ring, it must face down.

- All modern pistons of quality design utilize a 3-piece oil ring design with a center expansion ring and two outside rails.

- Older pistons use a thick cast scraper ring or not-so-old oil rings use a single cast scraper with relieving in the center portions for oil removal.

- Grinding the rails to attain correct endplay is acceptable and necessary but do not grind the center expander or allow it to overlap.

Ring gap is a combination of the ring expansion itself and the expansion of the cylinder bore. Therefore, the metallurgy of the cylinder is important in determining correct end gap. Axtell took the time to measure stock Evo and Twin Cam aluminum cylinder growth and found expansion rates up to a full .009” and slightly beyond at operating temperatures. 11 Cast iron or ductile cylinders the growth is much less.

Calculation of Extra Ring End Gap for Cylinder Expansion

e.g. aluminum cylinder .009” expansion

Cylinder expansion rate x Pi = extra ring gap

Example

.009” x 3.14159 = .0283”

Pi or $\pi$ is a mathematical constant. 3.14159.

It represents the ratio of a circle’s area to the square of its radius.

Calculation of extra ring gap is not necessary in most cases, as the manufacturer will give correct specifications for

the intended use.

However, another way to calculate ring gap is a cylinder expansion rate and cylinder bore estimate.

Aluminum Cylinder Top Ring Gap Calculator

e.g. CVO 110 aluminum cylinder with 4.000” bore diameter

Bore diameter x .004” = ring gap

4.00” x .004” = .016” ring gap

When using aluminum cylinders the minimum ring end gap is the way to go since the cylinders expand so much.

Cast Iron or Ductile Top Ring Calculator

e.g. CVO 110 aluminum cylinder with 4.000” bore diameter

Bore diameter x .006” = ring gap

4.00” x .006” = .024” ring gap

I repeat, when using aluminum cylinders the minimum ring end gap is the way to go since the cylinders expand so much. Cast iron or ductile cylinders have a smaller expansion rates so ring gaps should be beyond middle recommended figures towards the maximum.

Middle Ring Gap Calculator

e.g. Aluminum, cast iron or ductile cylinder, with 4.000” bore

diameter

Bore diameter x .004” = minimum ring gap

4.00” x .004” = .016” minimum ring gap

Bore diameter x .005” = maximum ring gap

4.00” x .005” = .020” = maximum ring gap

Middle rings are somewhat protected from excessive heat by the top ring that withstands the worst of cylinder combustion. Therefore the middle ring does not usually need the maximum ring gap settings because there will not be as much expansion as with the top piston ring.

Many mechanics and riders want their engine tolerances

set to minimum specifications but this is not always wise even though the engine will have proper break in. Therefore, larger ring gaps should not concern the mechanic because gaps tighten up considerably when the engine reaches operating temperatures. 12 Hence, a reason not to be too conservative with gapping rings.

In addition, a thinking mechanic will look at aggravating heat factors such as too-high rear cylinder AFR ratios with closed loop EFI or restrictive stock exhausts and stylish long exhaust systems that add substantially to engine heat. When there is heat aggravation, always use the higher end of recommended ring gaps.

With 3-piece oil rings, a center expander and outside rails, top and bottom, the corrugated expander ring ends will buttress together after installing the rails and fitment into the

cylinder. The expander as the name suggests provides the tension on the rails, which scrape the oil off the cylinder wall. Do not modify the expander, period! The rail end gap should fit between .015” and .050” gap. 13 This is a wide variance.

Use the principles above to determine best rail end gap for a particular engine such as cylinder metallurgy and aggravating heat factors, if any. When building an engine properly, it is necessary to think through every aspect of construction instead of daydreaming while blindly following specification tolerances. Modify tolerances to the specific needs of an engine as well as the type of riding use the engine will enjoy or suffer through.

I build engines according to customer personality. It is important for me to spend some time with a new customer to figure out his or her riding style. In simplistic terms will the intended rider be overly conservative and lug the operate it within prescribed break in procedures and once broken in stay

within proper rpm ranges in all gears and change oil regularly or beat it up?

The mechanic should modify the tolerances accordingly.

Adjusting Ring End Gap

Once again, never adjust the 3-piece oil-ring-center-expander.

Grind or file the top, middle or bottom rails oil ring rails from the outside in. Never file or grind from the inside out or the ring face may chip causing undesirable consequences in the running engine. Remove burrs after adjustment. 14

As always, cleanliness is next to godliness.

## Oil Ring Tension

Axtell recommends testing the oil tension by installing only the oil ring on its piston. Lightly oil the cylinder. Insert the piston into its cylinder. Use a fish scale to drag the piston through its bore at a constant speed. This measures the amount of drag generated by the ring tension.

Axtell testing has shown that maintaining oil control is possible from 9-lbs pull on a maximum output engine to 27-lbs drag for a high mileage touring application.

Again, rail-ring-end-gap is a determining factor in drag or pull if the tension expander is doing its job. 15

## Ring Orientation

Follow the instruction sheet for expander rail orientation as there are different methodologies basing on design differences. In general, the top rail will be around an inch to one side of the expander ring opening and the bottom rail around one-inch to the other side of the expander opening. The mechanic orients the ring gaps equidistant from each other to ensure compression does not blow downwards past the rings into the bottom end and to prevent oil making its way up into the combustion chamber.

Boring and Honing

We want a flat surface on which to base our work. If surfaces are warped then every measurement afterwards will suffer.

1. Face head and base gasket surfaces of a cylinder first before resizing.

2. Always use torque plates because they simulate actual install conditions on the engine. When cylinders are un-torqued during disassembly, the bore will spring to a different shape and not necessarily be round and concentric top-to-bottom. They may go out of round and taper.

At Heavy Duty Cycles (www.heavydutycycles.com), we bore and hone with torque plates to simulate actual engine operating conditions so that the installed cylinders will be round, perpendicular and concentric top-to-bottom.

3. Also, torque cylinders to specification with installed torque plates for checking piston sizing to cylinder bore to ensure correct tolerances.

4. Boring is superior to honing for piston fitment because the bore will be perpendicular to the gasket surfaces.

- Precision honing using fixed adjustable mandrels will create a round bore but it will not necessarily be perpendicular.

- Deglaze honing with deglazing centrifugal twirl balls or spring-loaded expander hones will not necessarily cut round or perpendicular.

- Furthermore, use the correct size torque plates. Do not use larger cylinder bore torque plates on smaller bores as the pressure of the plates will not have equitable distribution. Larger inner diameter torque plates can distort the base and head gasket faces outward thus pinching the formerly round bore inwards in the center.

After boring in this uneven situation, when torquing the cylinders during engine assembly, the now even application of pressure will leave a bore too wide in the middle tapering smaller to each cylinder face. Piston thrust surfaces and rings will suffer much trauma contending with these circumstances.

I mention this because torque plates are expensive and there are so many different sizes bores on Harley-Davidson's currently from 3.00" on earlier engines to present behemoth 4.500" big-inchers. I would have to go count the torque plates we have at Heavy Duty but I bet we have at least seven sets and our machinist Marvelous would have more.

When boring a cylinder, leave up to about .008" clearance before attaining piston fitment measurement. The reason is that correct fitment and cylinder wall finish takes numerous steps to fulfill.

Axtell uses 5-steps for honing with 5-increasing stone grits; 70, 180, 220, 280, and a cork wiper finish. 16

Of course, the honing finishes should be at a crosshatch to both allow break-in and to contain oil on the cylinder sleeve surface.

Wrist Pin Keepers and Teflon Buttons

Axtell uses two types of wrist pin keepers, .073” wire and Spiro locks. The wire lock has a groove on the bottom and has a notch cut in the piston to aid removal.

- The Spiro lock uses a .044” seating groove in combination with a conventional square bottom groove

without piston notching.

- Some pistons have a double Spiro lock system with a .088” groove to accommodate them.

- Spiro locks resemble a small flat-coiled spring. Only pull the spiral coil apart up to about a quarter-inch. If overstretching the lock, start over with a new one. Do not try to compress the natural diameter circle of the spiral lock for ease of installation. This will cause deformation of the lock and disturb correct radial preload.

- Gingerly place the end into the piston-pin-boss-groove and spiral the complete lock into the groove until the lock seats fully in the groove. This is not as easy as it may sound. It requires practice, dexterity, and above all, patience.

- The spiral lock should not move in its groove after

installation. If it can, remove and try another new one. Never re-use a spiral wrist pin lock!

- Some pistons do not use locks but have a Teflon button holding system where the wrist pin and buttons float with preset endplay.

Teflon buttons may add more stability, particularly as the piston changes axis at TDC and BDC. They do not add extra thrust surface area as the wrist pins and buttons house perpendicular to the sides between fore and aft thrust surfaces.

Axtell supplies different depth buttons to achieve correct wrist pin endplay of .020” to .060”. Endplay specification will depend on metallurgy, generated engine heat, and piston-to-cylinder tolerance.

I will comment on one factor, which is heat. Obviously, an engine operating with a closed loop EFI and Stoichiometric 14.7:1 air-fuel-ratio will generate more heat than previous lower AFR fuel ratio engines. Furthermore, the rear cylinder will run even hotter because of cooling airflow-shrouding conditions. Therefore, the rear cylinder wrist pins and dual Teflon buttons need more tolerance endplay with closed loop EFI air-fuel-ratios running well above 14.7:1 Stoichiometric AFR.

Compare this situation with a front cylinder on a carbureted engine running at around 14.0:1 AFR where endplay will be a tighter fit with less tolerance.

Once deciding on endplay, add this to the length of the wrist pin. Subtract this from the cylinder bore and divide by two. This will equal Teflon button depth.

Calculate Wrist Pin Teflon Button Depth

e.g. wrist pin 3.200”: desired endplay .040”: cylinder bore 4.000”

Wrist pin length + desirable endplay – cylinder bore ÷ 2 = button depth

Example

3.200” + .040” – 4.000” ÷ 2 = .380”

Balancing is necessary when changing piston weight. 17

If necessary, equalize piston weight by grinding or

polishing underneath the heavier piston, usually near the piston pin bosses, which are the meatiest portions. Compare new piston weight with the former pistons. If the weight is equal or very, slightly more on the new piston then everything is okay.

Furthermore, if having static balanced the engine previously, the various figures will allow calculations to bring the new pistons within the set balance factor.

Otherwise, new pistons require the static balancing to a balance factor of 60%. I will go into this advanced-mechanical subject further in Chapter V: Twin Cam

Vibration, subsection, Balancing: Reciprocating and Rotating Weight.

Axtell 110c.i Cylinder Gasketing

Axtell supplies the excellent Cometic gaskets (www.cometic.com), with their cylinders with which, I am going to replace the stock CVO cylinders.

Twin Cams use an O-ring to seal the cylinder to the engine casing deck that has proven very successful over the years. It combines with two OEM 11273 Viton dowel O-rings to seal oil in the oil return passage in each cylinder on its journey back to the timing compartment.

Axtell utilizes a Cometic coated-metal base gasket to replace the main H-D O-ring on the Axtell 110 cylinder base. With this kit and base gasket, Axtell still uses the two OEM 11273 O-rings for the base cylinder dowels. This base gasket has sealing ridges that mate with the O-rings around the two locating dowels, one of which also serves double duty as the

oil return passage. Another sealing ridge is evident on the inner diameter of the base gasket.

Do not use the OEM 11273 O-rings in conjunction with the CVO 110 or the Axtell multi-layer-system (MLS) head gaskets.

The CVO-110 MLS second-generation gasket that H-D hoped would rectify the oil-leaking situation is a three-in-one gasket. It has a metal center with a rubber Viton coated metal gasket on either side. There are two metal rivets, each on a gasket appendage opposite to each other on either side of the bore to hold the multi-layer gasket together.

MLS gaskets will not tolerate a rough or warped gasket surface.

There are three major problems with the CVO 110-rear MLS head gasket sealing correctly.

1. The head gasket surfaces that I have thus far observed are not always smooth enough to allow an MLS gasket to work properly.

2. Secondly, the CVO 110 rear head has a tendency to warp under the extreme heat of up to 16.0:1 air fuel ratios in the rear cylinder.

3. Thirdly, the shifty CVO 110-cylinder liner at the top of the cylinders creates a bump that once again, a MLS gasket cannot tolerate effectively.

Interestingly, the thick and fibrous TC88 head gasket style complete with a fire ring to seal compression into the cylinder along with OEM 11273 dowel O-rings for oil-return sealing will help answer the three concerns listed above. This style

gasket can accept a rougher gasket surface and can cope with more head warpage without leakage. The fire ring can manage with a shifty cylinder liner and lastly the OEM 11273-cylinder dowel O-rings will effectively seal oil within the oil return passage.

The H-D MLS gasket looks so identical to the Cometic (554-241), head gasket that I am going to say that they are. Therefore, the irony becomes that I will use the same MLS gasket that fails on the H-D CVO 110 cylinder to help successfully deal with the problem using an Axtell 110 cylinder that has a smoother gasket surface and a cylinder liner that does not move.

There is a similar but different head bolt tightening procedure.

Guess what, H-D's next new attempt after the failure of the

MLS gasket through no fault of its own is similar to the TC88 gasket! There is no part number for these gaskets at time of writing but markings on them include “IVAN-AC” and “This side up 4.000”.

In using the Cometic MLS gasket with Axtell cylinders, ensure the rivets holding the 3-piece gasket together do not pinch between cylinder and head. If so, remove the rivets but keep the same gasket configuration and relationship as before when riveted.

## Axtell Cylinder Installation Instructions

### Cleaning Cylinder Bores

The Axtell cylinders have a protective coating on the sleeves to protect from oxidation also known as rust.

1. Wash the cylinder liners thoroughly in warm soapy water using a firm plastic bristle brush.

2. Rinse cylinders and dry completely, then oil with a white paper towel, as rust will begin to appear on the liner surface. In this case, rust is an indicator of a great cleaning job.

3. If the paper towel shows evidence of dirt, repeat the washing procedure and do a more thorough job.

BDC Piston Clearancing

Mock up the piston and cylinder assemblies.

1. The notched piston installs in the rear cylinder, notch

forward to provide relief for the front piston at or near BDC.

2. The cylinder bottoms are stamped “F” and “R” for front and rear installation on the engine.

3. On the mockup, with pistons and wrist pins fitted in the connecting rod bushing small ends, raise the cylinders off the engine case deck far enough to look at the piston bottoms.

4. Rotate the engine slowly and observe the piston skirt interaction as they approach BDC, change axis and then direction. There must be minimum .060” clearance between piston skirts at their closest point. This is standard stroker and big bore building procedure. If not a minimum of .060” between the pistons, remove the pistons and clearance accordingly.

TDC Piston to Valve Clearancing

Cam lift, cam duration, intake and exhaust cam lobe separation angle (LSA), valve margin, head design and aftermarket milling of cylinder heads determines piston–to-valve clearancing.

Piston crown height (compression), crown design and bore diameter will reflect some of the above clearancing demands.

I mention this because; it is crucial to use the correct pistons for the above applications. Generally, manufacturers save the mechanic and consumer much of this legwork with their designing.

Read the instruction sheets! Different manufacturers will have different piston-to-valve clearance specifications

1. The Axtell piston with intake valve open at maximum lift must have a minimum .090” clearance between it and the piston at top dead center.

2. The exhaust valve must have minimum .060” clearance between it and the piston at TDC when fully open.

Do this procedure by using putty like Plastocine on the piston valve pockets or in the area of closest interaction between valves and piston at TDC when each valve is fully open.

1. Mock the top end onto the engine.

2. Rotate the engine slowly by hand.

3. If feeling any resistance, stop immediately, and disassemble to determine where the valve contacted the piston crown and relieve.

4. After successful rotation, disassemble and measure the thickness of the putty where the valve and piston made contact with it.

5. If meeting the above tolerances, all is well and final assembly may proceed after weighing pistons to ensure equal weighting between the two.

6. If the minimum tolerances are not met relieve the valve pockets to attain minimum tolerance, reassemble, and re-measure until satisfied.

7. Weigh the pistons and remove material underneath in the thickest area to equalize weight for balancing purposes to

prevent unwanted vibration.

Squish Clearancing

After this, slide the cylinders down and seat them on the engine case deck. Rotate engine to place pistons at TDC. Check squish clearancing, which is how close, even or above the head gasket deck one wishes for best turbulence of air and gas. This subject is complex, depends on piston dome shape and other factors that require a competent performance mechanic.

The results of taking the time to engineer squish clearancing will pay off with first-and-foremost more power.

A squish band is the area or areas where the piston comes in close proximity to the chamber as it changes axis through

top dead center. Air and gas caught in this space squeezes out from the squish band at high velocity generating turbulence in the combustion chamber. Turbulence in the combustion chamber promotes better atomization and less separation of the fuel and air, encouraging it to burn.

A proper squish clearance makes the engine both more efficient and less prone to

detonation because the air and gas are powerfully mixed. Moreover, fuel mileage will increase because of superior mixing of fuel, as well as the aforementioned reduction in spark knock.

If confident in your mechanical ability, squish clearance should be set tight to about .030”. Tighter than this risks piston to head, contact. A looser squish clearance progressively begins reducing the chamber turbulence that is

so necessary to a faster and more efficient engine. Stock engines usually has .050” squish or above.

One style Screamin’ Eagle head, a hemi-version has an angled squish band where the squish band is at an angle from the horizontal. If not looking closely, one might think it a simple and thus inefficient hemi-head. However, the piston hemi-dome has a similarly angled squish surface to mate with the chambers squish band.

More common in stock H-D engines like the Evolution and the Twin Cam is a zero degree or flat squish band, which is the same as the angles squish band except the air/gas squeezes out horizontally.

Axtell recommends a compression squish of .030” to .050”. Obviously, .030” gives superior squish while .050” is for the

conservative. In closing, too little squish between piston top and the flat squish band surrounding the combustion chamber will result in less turbulence, poorer mix of gas and air and thus wasted power potential. Too much and parts interference may have disastrous results apart from performance considerations. Gasket thickness can have a large influence on squish after the piston relation to deck height, of course.

## Lifter Cover Clearancing

The lifter covers may need slight relieving if they contact the beefier Axtell cylinder bases.

## Cylinder Assembly

Before assembly, liberally coat all internal parts with high quality 20W-50 oil to assist in startup prevention of oil starvation while waiting for pumped engine oil to arrive.

1. Do not use O-rings around the cylinder-to-head dowel holes. The gasket has embossing around the dowel area and the dowel holes are smaller. The head and base gasket surfaces of the Axtell cylinders is impeccably smooth, which is great because imperfections on the gasket surface will cause failure with the high tech MLS gaskets.

2. Clean the head bolt threads and apply a light coating of light machine oil on them and under the head bolt underside flange. Do this to have accurate uncompromised torque specifications values. Dirt and other resistance to threading in the bolts will alter effective torque ratings.

3. The two shorter head bolts locate on the spark plug side of the head-to-cylinder.

4. Therefore, the two long bolts locate on the pushrod side of the engine.

5. Alternatively, snug the head bolts finger tight. In sequence, tighten the rear short bolt, then the front short bolt, then the rear long bolt and lastly the front long bolt to 14 ft-lbs.

6. In the same sequence, now tighten the four head bolts to 22 ft-lbs.

7. Then tighten in sequence to 35 ft-lbs and finally to 42 ft-lbs.

Harley-Davidson Torque Specifications

Harley-Davidson advises the same tightening sequence as Axtell.

1. The oiling procedure naturally substitutes with H-D 20W-50 oil as opposed to any quality 20W-50 oil.

2. The H-D bolts torque first with finger tightness, then 120-144 in-lbs.

3. Then tighten to 15-17 ft-lbs.

4. Finally, take a marker and stain the position of the bolt to the head by marking a washable line across the bolt head onto the head surface.

5. Now retighten in the same sequence a full quarter turn or 90-degrees.

# Engine Tuning

There are three major considerations to examine which are air-fuel ratio, ignition timing, and oil pressure.

AFR must be rich. If adding an aftermarket fuel modifier, it will probably be on the rich side but checking is always necessary. A lean air-fuel-mixture leads to excess heat, which is especially injurious to new pistons and fitted cylinders. Leanness will lead to detonation, the great engine destroyer.

While the CVO 110 is fuel-injected, carburetion settings for a new engine will add clarity to this subject. Install a larger low-speed jet to provide more fuel at idle for cool running. Main jet changing is not so important because they control fuel at highway speeds beyond the rpm range for correct engine break-in.

The bottom line is that new performance engines require enrichment if in doubt at all. Fuel cools while air heats.

Advancing spark ignition too much will also increase deadly heat. Start at 32-degrees full advance for initial startup and tuning.

There is lots of time to dial in ignition timing and AFR but it is paramount to begin in safe fuel and spark ranges.

Ensure there is oil pressure on startup. Axtell kits require at least 20-psi oil pressure at 3000 rpm. Always check but Twin Cams usually have around 35-psi at 2000 rpm.

## Break-in

Do not believe the crowd that advocates full tilt-and-boogey right out of the gate. Their attitude is if the engine is no good it will blow right away. If it holds together, it is a good engine. All I have to say is bullshit!

Any competent mechanic can disassemble an engine and read it to analyze why the engine self-destructed…say goodbye to warranty.

1. The proper break-in procedure is to heat-cycle the engine by starting the motor with a heavy fan swathing cool air over the engine to dissipate heat.

2. Initially, fire the engine and allow running for only 30-seconds.

3. Allow the engine to cool to room temperature.

4. Restart the engine; allow running for 60-seconds, and then cooling down.

5. Repeat this procedure adding 30-seconds each time until reaching a total of four minutes or 240-seconds or 8-cycles of heat cycling.

6. Change the oil and filter while engine is hot. Cut the filter open to observe particulate, which will indicate any problems as well as success.

7. Now, ride the bike for 1000-miles keeping the engine under 3500 rpm for 300-miles and less than 4000 rpm for the remainder.

After this, still respect the engine until warm. Gradually increase rpm levels. Do not lug the engine or put undue stress

on it such as towing a trailer. In fact, never lug an engine whether stock or performanced. 18

## Conclusion

In conclusion, I am taking no chances with Chris's CVO 110.

The two previous dealer rebuilds have been very ineffective.

I have lowered the AFR from Stoichiometric to around 14.2:1 or lower depending on circumstance with an input fuel modifier, Terminal Velocity. My very close but second choice for this application is an output fuel modifier, the Power Commander, which also has an ignition component unlike most of the other output modifiers. I chose Terminal Velocity because it is self-tuning based on real time oxygen sensor

information and will adapt to changing conditions.

There are many excellent fuel modifiers on the market and many not-so-good ones. Ignition and fuel modifier technology for Harley-Davidson is constantly changing. This is my current choice. Next month, an improved system may come along.

The key is to have trust in a very competent mechanic and abide by his or her decisions. If that person cannot explain comprehensive reasons for using a specific part over another, find a different adviser.

I have spent much time analyzing the reasons there is persistent and frequent CVO 110-rear head-gasket-oil-leaks without a compression leak.

It is obvious that cylinder redesign for the stock CVO 110-

cylinders is a priority for the Motor Company especially utilizing spiny-lock high-pressure-cast construction up to within the top cylinder deck much the same as used in the TC88 workhorse cylinders.

The aftermarket serves the H-D rider well once again, as I am able to use the venerable Axtell cylinders and pistons to help ensure Chris can ride problem free on his CVO 110 after the third rebuild.

Please see Chapter XVI: 2009 Model Changes for CVO 110 gasket, EITMS and engine changes.

1 Rice, Jerry. Memphis, TN.  Email to Heavy Duty Cycles Limited, Friday, August 17, 2007 2:36 PM. Subject: head gasket leak

2 AIM reader who wishes to remain anonymous. Marietta, GA.

Email to AIMag.com), Monday, December 17, 2007 2:44 PM. Subject: Head Gasket Leaks on H-D 110's.

3 Email to AIMag.com), Guelfo, David. Sunday, January 20, 2008 9:52 AM. Subject: Re: SE

4 Harley-Davidson. 2006 Harley-Davidson Genuine Motor Accessories and Genuine Motor Parts. Page 442, E.

5 Wood, Bob. Wood Performance Carburetors. Part of a telephone conversation early 2007.

6 Axtell Sales. Website, (www.axtellsales.com). Products, Twin Cam, Twin Cam Cylinder Kits, Sku, 200-215S.

7 Axtell Dyno Master Performance Products Home of Real

World Power. Twin Cam Instructions. Pages 5-11.

8 Ibid.

9 JE Pistons, Sportsman Racing Products. Instruction sheet 4032

10 Axtell Dyno Master Performance Products Home of Real World Power. Twin Cam Instructions. Pages 5-11.

11 Ibid.

12 Ibid.

13 Ibid.

14 Ibid.

15 Ibid.

16 Ibid.

17 Ibid.

18 Ibid.

# Chapter IV

## Cams, Cams, Cams

### The Heart

A Harley engine equates to an air pump; air draws in, compresses, burns in conjunction with fuel, expands and finally expels the pump only to repeat these four cycles repeatedly.

How efficiently the air moves through these cycles determines engine power. The camshaft(s) four lobes open and close these four cycles allowing air to enter, compress, expand and exit.

In general, duration, or how long a valve holds open before, during and after its theoretical 180-degree cycle determines rpm potential of an engine and thus influences the power band. An increase in duration will increase the rpm levels in which the power band operates.

Lift is the distance a valve opens. Increase in lift will lower rpm levels that the power band operates. Increased lift increases torque.

These gross simplifications serve to show some effects of cam profile choice. There are many other factors affecting a cam profile, which I will discuss such as overlap, advance-retard cam timing, lobe center angles and the lobe separation angle.

Power Band

A power band is the operational area in which an engine creates power. A power band is approximately 3000 rpm wide.

Unfortunately, we cannot have good peak power at all rpm ranges. A pushrod-operated engine has limited rpm capabilities right out of the gate because mechanical efficiencies can only move a cumbersome valve train so fast. Therefore, the Harley engine is limited in rpm capabilities. Try to move a valve train beyond its capabilities and pushrods may bend, valves may float and bounce off their valve seats and valve springs may oscillate to name a few possible destructive results.

Hence, overhead cams as found in the V-Rod can rev much higher as the valves actuate directly via the cam lobe.

The power band can have one of three areas of strength; low from idle to about 3500 rpm, mid-range from around 2000 rpm to 4500 rpm or high from about 3500 rpm to redline, which will be somewhere around 5500 rpm and with about 6500 rpm being maximum.

Now, it becomes readily apparent how important cam choice is. Choosing a high powerband cam profile for downtown city riding will not be enjoyable.

Unabashedly Bob Wood of Wood Performance (www.woodcarbs.com), states repeatedly that he designs torque, torque and more torque into his camshafts. Thus, many of his cam profiles feature lift over duration as well as a few other factors I will explore. The great thing about Bob is that he will personally steer you into the proper cam profile

for your bike, performance objectives and riding style...and the education you receive along the way is free.

## Design Evolution

The Environmental Protection Agency (EPA) dictates that the noise emanating from 1999 and earlier Harley-Davidson Big Twin gears driving the camshaft whose 4-cam lobes operated their respective four valve trains was unacceptable. This simple progression to a quieter environment drove the design change to cam timing chains.

For many years, Harley-Davidson valiantly resisted a change to our beloved traditional engine shape and design by spending much time and effort to closing up the tolerances between mating gears driving the valve train. However, the physics of expansion and contraction with heating up to engine conditions and cooling down at rest finally defeated

the Factory's attempts. The heat-expansion of gear-metal determines that the final tolerances of gear-to-gear fitment must occur at operating temperatures. Too tight fitment results in a high pitch whine, offensive to all. Too loose fitment dominoes into noisy gear clatter, not so offensive to traditional Harley riders because that is all they have known but very objectionable to the EPA.

Operating heat fitment to eliminate whine or clatter necessitated some clatter with a cold engine that gradually reduced during warm-up. These gears were of the spur or straight-cut type and unlike helical gears that are always in contact and thus much quieter. Gear lash clatter is the result of matching spur gears no matter how close the fitment. Of course, precise fitment within the bounds of physics results in lesser but still audible noise. Spur gears are not in constant contact. This leads to the condition called backlash.

An American Iron Magazine reader named Andrew Bispham, who is obviously a knowledgeable person, emailed me his take on backlash, “Here is a small tip which can be used to quieten spur gears. One of the causes of noise from spur gears is the line contact as each tooth leaves its partner. This line contact results in a perceptible “click” as the teeth separate. As speed increases this becomes a whine. This doesn’t happen with helical gears as they have a rolling point contact along the teeth and exit at the end, hence quieter gears. A trick to replicate this on spur gears is to crown each tooth with a radius (i.e. make each gear slightly barrel shaped, with the ends of each tooth .004” lower than the middle.) This results in the teeth leaving each other in a point contact in the middle of each tooth rather than a line. This eliminates the “click” and results in quieter gears. They will never be as quiet as helical gears but it’s a cheap solution with no end thrust complications. 1

Bob Wood uses a similar grinding process on his gear drive spur gears to what Andrew describes.

Harley-Davidson has been using a technology for over a decade now with its spur cut transmission gears for quietening purposes that involve grinding high spots off the gear faces after the hardening process.

The thrust complications Andrew refers to are the action of mating helical gears thrusting in the direction of forward rotation. There are different ways to deal with this. Bob Wood of Wood Performance resolves this thrust issue with his optional helical gear design for his cam gears on the inboard side of the cam support plate.

This prompts the obvious question of why H-D did not change to helical full-contact quiet gearing in their timing

chests to drive the valve trains. You got me?

As I have said, Bob Wood has this option with his Knight Prowler Advanced Belt Drive System for the actual mating cam drive gears on the inboard side of the cam support plate.

This combination of a carbon fiber belt driving the outboard cam support plate powering cam drive pulleys with inboard connecting helical cam gears has to be a winner with anyone concerned about EPA-noise restrictions. I suspect and hope that bigger companies are going to pick up or license the Knight Prowler Advanced Belt Drive System for this reason only…and then the bonuses are the exquisite cam cover with Bob's hi-torque cams on the inside.

To me, the noise is not a big deal with spur gear driven valve trains and transmissions although I know I will have to conform like everyone else.

What attracts me to Bob Wood's belt driven camshafts is the sheer beauty of the system. Ultimate cool is the only way I can describe it. Then there is the performance aspect. Wood Cams hi-lift, tight LSA, and resultant hi-torque cam profile specifications blast your bike off the line and keep the power cooking up the rpm range until shift-time when the grunt power starts all over again.

The Advanced Belt Drive System is the culmination of progression away from the stock cam chain drives and beyond the aftermarket improved gear drives.

I now ask the reader's indulgence and patience as I work through a bunch of math that will eventually tie together to simplify a complex subject. Bob Wood will help along the way as he could probably simplify nuclear physics with his down-

home folksy ways.

## Cam Lift and Valve Lift

Cam lift and valve lift are easy to understand.

To many, the terminology of cam lift and valve lift are synonymous although their interchangeable use is not correct. Cam and valve lift have closely related meanings but are quite different. The lift of the cam dictates the lift of the valve. The valve lifts off its mating valve seat opening into the combustion chamber.

Usually the measurement of cam lift is vertically from .053” rotation off the heel or lowest part of the elliptical cam lobe to the highest point. The reason for this starting place for measurement is that the valve does not really start to open until the cam rotates .053” off its heel.

The lift of cam lobe and valve is different on all Big Twins going back to the 1936 to 1947 Knucklehead when the two were equal. Cam lift is less than the valve lift going back to the 1948 introduction of the Panhead although valve lift is a direct result of the cam lift as they interconnect in the valve train.

Valve Lift

Enthusiasts and mechanics alike normally refer to valve lift, which is how far the valve leaves its mating valve seat and extends into the combustion chamber.

Cam lobe duration is how long the valve remains off its seat.

The intake valve opens to its dictated lift off its valve seat to allow the fresh intake charge into the chamber. The exhaust valve also opens to its instructed lift off its valve seat to allow burned exhaust gases to expel out of the chamber.

A valve train's cam lobe rotates to its highest point forcing its mating valve to push open. There are really four separate valve trains, each servicing its mating valve in a two-cylinder, four-valve engine like a Harley-Davidson. In practice, mechanics refer to the four separate valve trains actuated off the four cam lobes simply as the valve train.

As the cam lobe rotates to its lowest point, the valve forcefully returns to its valve seat. It is not the cam-lobe forcing the valve to close. The cam lobe merely allows the valve to close.

The valve's compressed valve spring(s) push up to force the valve closed at the rate the descending cam lobe allows. Two cylindrical shaped valve springs (18201-83 outer and 18202-83 inner), one inside the other hold each valve closed on early Twin Cams up until 2003. This spring set is one of the few parts carried over from the 1984-1999 Evolution motor into the Twin Cam engine. A single beehive shaped valve spring (18245-02), operates the closing of each valve on later Twin Cams from 2004 on. I will discuss the dynamics of valve springs and most other engine parts in Volume VI: How it All Valve springs are a deep but easily understood subject.

Cam Lift

The measurement of valve lift is always greater than that of cam lift on all H-D Big Twins from 1948 to present.

The 1936-1947 Knucklehead has a rocker arm ratio of

1.000. Therefore, cam lift and valve lift are the same on the Knucks.

Cam lobe lift measurement is very precise.

Valve lift is not so precise because of variables in the valve train. It is usually accurate but not as accurate as cam lift. Variables such as pushrod angularity, pushrod length, valve angles, lash between parts too loose and if the rocker arms themselves are the exact ratio advertised as they may be off a few thousandths-of-an-inch. This is not a big deal on a street engine but is one of many crucial details on a race engine. Real-life valve lift can be measured on a mocked up valve train without the upper rocker box cover using a dial indicator micrometer that measures actual lift opening at the valve stem.

Cam lift measures from the lowest point on the cam base circle to the highest point (centerline), of the elliptical cam lobe. Complicating matters a little is that in practice we usually take these figures not at the bottom lowest point of the cam lobe but at .053” off this point. Again, the reasoning is that nothing much is going on with valve train movement until this point in the cam lobe rotation.

Rocker Arm Ratio

The cam lobe lift multiplied by the rocker arm ratio determines actual valve lift.

Rocker arm ratios in all Harley-Davidson engines are fixed and do not adjust. I have seen aftermarket rocker arms that adjust but they are tricky to say the least requiring a very astute and patient mechanic to play with them.

The pushrod actuating side of the rocker arm is closer to the rocker arm's pivot point than that on the actuating valve side. This creates an increase in lift at the valve over that of the dictating cam lobe lift. I will go into this in more detail later.

Evolution and Twin Cam rocker arm ratios of 1.625:1 are the same. Shovelheads use a rocker arm ratio of 1.420:1 and Panheads of 1.500:1 and Knuckleheads are 1.000.

What you see is what you get with the Knucklehead 1936-1947. The cam lobe lift and valve lift are equal with no multiplication factor of pivotal rocker arm offset.

Interestingly Panheads and Shovelheads up 1970 used the same cam lift on cams that were interchangeable in both engines. However, the valve lifts on either engine were

dramatically different because of differing rocker arm ratios.

Manufacturers and enthusiasts alike usually refer to the lift of a cam using the valve lift, which is the cam lobe lift multiplied by the rocker arm ratio. For example, all 2007 TC96 engines use a valve lift of .471” on the intake and .474” on the exhaust.

## Coordinating Head Flow and Lift

As a passing note, when head designing engineers and flow bench technicians measure head cubic-feet-per-minute (cfm) airflow, they do it at differing valve lifts. The exercise becomes how much air will flow when the valve is open at say, .550” lift.

Head port design as well as with porting and polishing existing heads, the cfm flow is at different rates for differing

valve lift openings. Therefore, port design determines optimum lift for the cam. As a simplification, what is the point in installing .650” lift cams in an engine, whose heads flow maximum cubic-feet-per-minute airflow at .550” valve lift?

Stock TC88 or TC96 heads can hardly handle cams with either the lifts compared above when they match cams with lifts of anywhere from .471” to .484”. This is another reason why headwork is crucial in performancing the Twin Cam.

## Calculating Cam Lobe Lift

(For TC96 intake cam specifications .471” valve lift and a rocker arm ratio of 1.625)

Valve lift ÷ rocker arm ratio = Cam lobe lift

.471 ÷ 1.625 = .290 cam lobe lift

Calculating Valve Lift

Cam lobe lift x rocker arm ratio = valve lift

.290 x 1.625 = .471” valve lift

Calculating Rocker Arm Ratio

Valve lift ÷ cam lobe lift = rocker arm ratio

.471 ÷ .290 = 1.625

Rocker Arms

The chrome wafer pictured on the upper left is an inside view of a stripped Twin Cam lower rocker box. An upper wafer (not shown), bolts to the lower one to complete what we see from the outside on the motorcycle engine. Evolutions have a system of three wafers, a lower, middle and upper. The lower Evo wafer incorporates integral rocker arm support stanchions that house the rocker arms and their shafts. Thus, the Evo system is load bearing, which is prone to transferring noise and gasket leaks. Reduction of internal transferring noise and leaking oil are two of the major reasons for Twin Cam development. The two Twin Cam wafers act as sealing covers only and are not load bearing. The Twin Cam is the first overhead valve Harley-Davidson since 1936 except the Panhead 1948-1965 that uses the rocker boxes as covers only. Tin pan pressed sheet metal rockers did a wonderful job of transferring all the Panhead valve train noise to the outside world. The lower left Twin Cam rocker arm stanchion support

plate with installed rocker arms and breather assembly is separate from the lower wafer. It bolts through the lower rocker cover to the head thus taking its support from there. The breather apparatus in the center also bolts through the support plate into the rigid head casting. The two rocker arms, on either side; one activating the intake valve and the other, the exhaust valve have a rocker arm shaft that slides through the stanchion housing, through the center of the rocker arm and into the other stanchion. The rocker arm rotates on the rocker arm shaft actuated on the right extension by a pushrod (not shown), following the rising elliptical cam lobe from below. As the cam lobe (not shown), rotates after peak lift downwards, the rocker arm rotates back to a neutral centering position as compressed valve spring(s) (not shown), push up on the rocker arm rubbing pad underneath the rocker arm extension to the left. The positions of the two rocker arm extensions in relation to each other determines rocker arm ratio. If the rocker arm extensions are on the same plane, the ratio will be 1:1 like those on the 1936-1947 Knuckleheads.

1984-present Evos (XL and Big Twin) and Twin Cams use a rocker arm ratio of 1.625 where the arms offset to each other by this ratio. The cam lobe lift multiplies by this ratio to get the valve lift, which is how far the valve moves off its seat into the combustion chamber. The center breathing apparatus separates gaseous vapour oil from air. Air expels outside the head into the air breather assembly while the oil drains back down into the timing chest. I extensively cover the breather apparatus in Volume I: The Twin Chapter IV: Harley Oil Leaks: Marking its Spot and in this Volume Chapter XII: Cubic Inches: Strokers and Big subsection, The Many Cures for Puking Oil. Bottom right pictures two rocker arm shafts that allow the rocker arms to rotate back-and-forth in the support plate stanchions. Upper right are two rocker arms. The upper one is a stock 1.625 ratio 1984-present rocker arm. The upper arm extension shows the rubbing pad that rubs across the top of the valve stem, pushing the valve sideways against its supporting valve guide and downwards opening the valve into the combustion chamber. The lower arm extension shows the

pocket that houses the upper pushrod ball. The lower rocker arm is an Ultima 1.675 rocker arm ratio. We cannot see the differences in rocker arm ratios because it is only .050” greater than the stock rocker arm. However, this ratio increase will open the valve anywhere from about .015” to .025” more than a stock rocker arm depending on the operating cam lobe lift. For example, the 1.625 stock ratios will take a .585G S&S camshaft to .585” lift. The 1.675 rocker arms will extrapolate the valve lift of the 585G cams to .603” valve lift. The pushrod pocket is the same as stock but the roller on the upper extension, rolls across the top of the valve stem eliminating much side loading on the valve guide and some frictional horsepower loss.

How to Figure out an Unidentified Cam’s Lift

Bob Wood, the superb teacher who reduces the complex to simple understanding, taught me this one.

Suppose, the reader picks up a cam at a swap meet or wherever but has no idea of the lift. Measure the base circle, which is the widest part side-to-side of the cam. I picked a cam lying around my shop. The base circle measures about 1.305". Now measure the centerline of the cam from the highest point of the cam lobe to the base circle top-to-bottom. The cam I am using measures at about 1.595". Subtract the lower figure (base circle), from the bigger one (top-to-bottom) to get the cam lobe lift. In this case (1.595" – 1.305" = .290"), the cam lobe lift is .290".

Now multiply the cam lobe lift times the Twin Cam rocker arm ratio of 1.625 to arrive at the valve lift of .471" (.290" x 1.625 = .471"), which is a stock intake cam used in early TC96 engines.

This cam may be out of my personal ride.

## Potential High Lift Cam Lobe Interference

I am briefly going to describe a potential problem that cam manufacturers eliminate or re-design during the design process.

This potential high lift cam-lobe interference as the dual cams rotate side-by-side is for diehard gearhead reading interest. If not a diehard, the reader can skip to the next section.

There are two different methodologies; one in use for timing chain drive cam sets and the other for gear drive cam sets.

### Chain Drives

Cam designers like Bobby Wood initially resolves high-lift cam-lobe interference issues in the very beginning of the Twin Cam's introduction as exampled with his TW9F .650 lift chain drive cam sets.

The Twin Cam camshaft lobes rotate beside each other creating a potential interference problem when increasing cam-lobe lift. The front and rear mating cam lobes on the two respective rotating cams come close to each other for an instant every 360-degrees when the highest point of the respective mating lobes pass each other.

Bobby Wood uses up to .650" valve lift cams, which have a .400 cam lobe lift in his Twin Cam chain drive applications by changing the cam lobe base-circle dimensions.

For example if a stock cam base circle is 1.305”,

- Bobby decreases his base cam circles to 1.190”, which allows an increase in lift up to .650” lift without interference problems (1.305” – 1.190” = .115” extra clearance).

- If the stock cam valve lift is .473 (.291” cam lobe lift), we can add this extra .115” clearance created by Bobby’s design to allow a cam lobe lift of (.115” + .291” = .406”).

- The cam lobe lift for .650” valve lift cams is .400”.

- To get the valve lift we multiply the cam lobe lift by the rocker arm ratio, which is 1.625 to get the valve lift (.406 x 1.625 = .660” valve lift).

Therefore, .650” valve lift cams now fit in a chain drive by reducing the base circle measurement of a stock TC88 cam

from 1.305” to 1.190”).

Gear Drive Cams

The passing cam lobes interference distance is approximately .110” on stock cams. This exact measurement is slightly different for every engine because production tolerances vary from engine to engine.

It is common practice to leave about which translates into .0625”, which we round off to .060” in practice, clearance between potentially interfering moving parts.

If a stock Twin Cam camshaft has .473” valve lift (some have .471” or .474” or .484”valve lifts), then a Knight Prowler .650” lift cam as I am now using means I have increased the valve lift by .177” (.650 - .473 = .177”).

However, we do not use the valve lifts for this calculation since they are a result of multiplying the actual cam lobe lift by the rocker arm ratio. We must use the cam lobe lifts, which for a stock .473” valve lift cam is .291”

The Wood Knight Prowler gear drive cams in my 2007 Streetglide are .650” valve lift, which translates into .400” cam lobe lift. If our clearance between stock mating lobes is approximately .110”, then we subtract .060” as a safety clearance factor. Now it is readily apparent that we can only raise the gear-drive cam lift safely by .050”. Any added lift and we breach the safety clearance factor.

The difference in cam lobe lift between the stock cam example and the Wood gear drive cams is .109” (.400 - .291 = .109”). This .109” figure is still more than double the safe limit increase of .050”.

The clearance grinding of the inside of the timing chest on the older Twin Cams would unnecessarily be too much. Newer Twin Cam engine casings such as my 2007 Streetglide have slightly different castings with more liberal tolerances so no clearance grinding for cam lobe interference was necessary for these particular lift camshafts.

For an actual clearance calculation, one would need the actual cam lifts, which are less than the valve lifts.

The correct formula is:

Calculation of Cam Lobe Interference

(Using .650” valve lift cams with .400” cam lobe lift)

The following applies to stock chain drives.

Performance cam lobe lift - Stock cam lift + .060 =

interference clearance required

.400” - .291” + .060” = .169”

After measuring the distance between mating cam lobes with stock cams, which will be in the neighborhood of .110” it becomes impossible to use a cam-set with such high lift unless the manufacturer has compensated with reduced base circle dimensions.

The highest possible bolt-in camshaft using stock cam lobe positioning designed by Harley-Davidson engineers is easily determined with the following formula.

# Calculation of Cam Lobe Performance Lift Increase

(Assume that the cam lobe interference is .110”. However, measure each application as this figure may vary because of differing production tolerances.)

Clearance between stock cams - .060” = possible performance increase in cam lobe lift over stock cam lobe lift of .291” (.473 valve lift).

.110” - .060” = .050” additional cam lobe lift

Therefore, we add the possible lift increase of .050” to the stock cam-lobe lift of .291”, which is .341”. We can install a gear drive cam with a lobe lift of .341” and still have a safe

interference cushion of .060”. To calculate the valve lift, which most riders and mechanics identify performance cams by; multiply the cam lobe lift times the rocker arm ratio that is 1.625 for all Twin Cams. Therefore, (.341 x 1.625 = .554”), a cam with a valve lift of .554” or less will fit in a stock Twin Cam using Harley-Davidson engineered cam lobe rotational positioning.

Okay, so there are probably a hundred other hi-lift chain driven performance cams over .554” valve lift in the marketplace will not fit into a Twin Cam engine using H-D cam design lobe placement specifications unless of course there is a reduction in the stock cam base circles of 1.305”.

If reduced by up to .115” down to a base circle dimension of 1.190” we can add that reduction figure to our previous maximum lift without an interference chain-drive problem. In practice, .650” to .660” is the safe maximum valve lift for a chain drive using the reduced base circle methodology with

chain drive cams.

Moreover, of course, the venerable S&S Cycle, Inc. was the first to make the change allowing the use of bigger camshafts in gear drive camshafts in the Twin Cam engine.

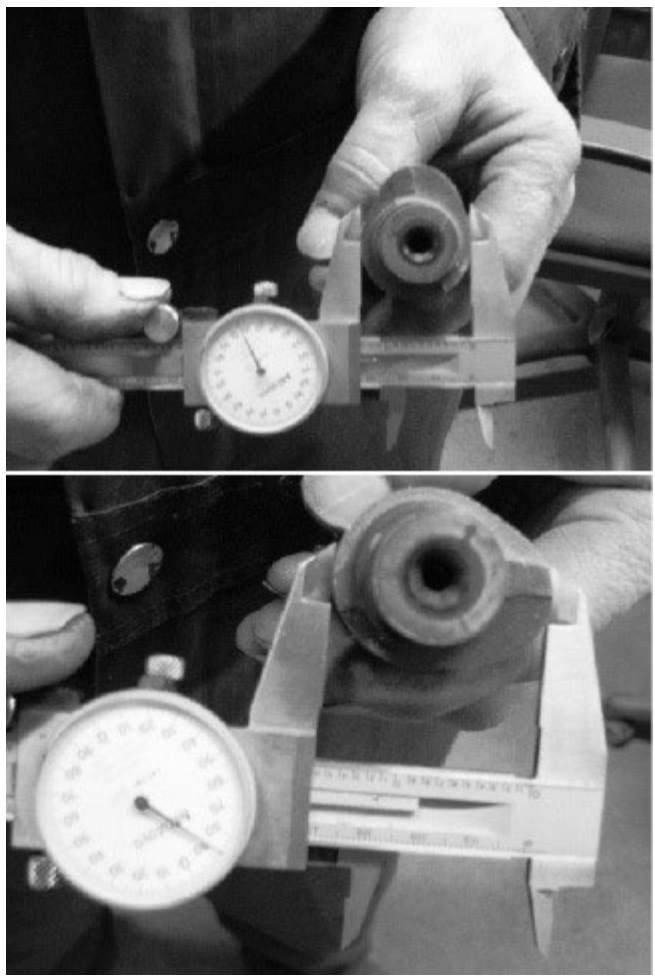
Reversing Rotation of Rear Cam for Clearance

To fix this potential interference situation for the hi-lift gear drive cams, S&S reversed the rotational direction of the rear camshaft from its normal rotation. Twin Cam cams rotate clockwise. The gear-drive rear cam now rotates counterclockwise while the front cam still rotates clockwise. Now, the mating cam lobe high points never come close to each other.

The front cam lobes are pointing in the same direction

with minor variances for performance timing with all hi-lift gear-drive cam manufacturers. However, by the time we take the gear driven reversal into account, the valves open and close in the proper timing sequence of the four engine strokes; intake, compression, power, and exhaust. With cam timing, lift and duration variances designed to produce power, the reversed lobes essentially open and close the valves in the same order and timing as the stock chain driven TC88 or TC96 cams.

Calculating Lift of an Unidentified Cam



Bobby Wood reduces the complex to simplicity. I have tried to measure a cam's lift so many times without success. First, it never occurred to me that cam lobe lift is much less than the lift at the valve because of rocker arm ratio extrapolation. I know, I should have thought about this immediately but for some reason, I did not. The minimum any Twin Cam, cam lobe will measure is 1.595" in height. Well this figure means nothing…in the beginning. How could it? We all think in terms of cam lifts being under .650" or much less, but nowhere do we think of lift as being over an inch-and-a-half! Therefore, an amateurish attempt in attempting to determine lift with one simple measurement becomes absurd. Suppose, the reader picks up a cam at a swap meet or wherever but has no idea of the lift. This is so simple, I felt dumb but Bobby Wood is very gracious in his teachings. The upper picture shows the measuring of the base circle of a cam lobe, side-to-side with a Vernier calliper. Measure the base circle, which is the widest part side-to-side of the cam. I picked a cam lying around my shop. The base circle measures about 1.305". The

lower picture shows the measuring of the cam centerline. Measure the centerline of the cam from the highest point of the cam lobe to the base circle top-to-bottom. The cam I am using measures at about 1.595". Simply, subtract the lower figure (base circle), from the bigger one (top-to-bottom) to get the cam lobe lift. In this case (1.595" – 1.305" = .290"), the cam lobe lift is .290". Now multiply the cam lobe lift times the Twin Cam rocker arm ratio of 1.625 to arrive at the valve lift (.290" x 1.625 = .471"), of .471", which is a stock intake cam used in early TC96 engines. This cam may be out of my personal current ride. Remember, I said the minimum any Twin Cam, cam lobe will measure, and top to bottom is 1.595"? Now, the reader knows why. The smallest stock Twin Cam valve lift is .471" with others being .473" and .484" over the years.

Increasing Lift Relative to Duration

As I have said before, many cam profiles other than Bob's

move the power band up too high in the rpm range for my personal liking.

High rpm riding is fun but not always practical.

I want grunt power in the lower rpm ranges, where I use my bike the most.

Do not worry, there will still be lots of power in the top end since we have probably put higher flow aftermarket heads on or at least ported and polished the stock heads.

The trick here is to increase the rocker arm ratio from the stock 1.625 for Twin Cams and 1984-1999 Evolutions, 1.420 for 1965 to 1984 Shovelheads, 1.500 for 1948-1965 Panheads, and last but not least 1.000 for 1936 to 1947 Knuckleheads.

Two increased rocker arm ratios that come to mind are a milder Ultima 1.675 and American made Crane Cams 1.750 ratio.

Suppose we take a TC96 intake cam-lobe lift of .290" whose valve lift becomes .471" after multiplying by the stock 1.625 rocker arm ratios. The TC96 breaks with previous Twin Cam tradition in using the same cam lobe lift and therefore valve lift on both intake and exhaust. I am using the intake as an example because the exhaust cam-lobe lift is .292" and therefore valve lift multiplies to .474".

Stock 1.625

TC96 valve lift is .290" cam lobe lift x rocker arm ratio of

1.625 = .471" valve lift

Ultima 1.675

Stock TC96 valve lift is .290” cam lobe lift x rocker arm ratio of 1.675 = .488” valve lift

Crane 1.750

Stock TC96 valve lift is .290” cam lobe lift x rocker arm ratio of 1.750 = .508” valve lift

Longer ratio rocker arms typically move the power band down increasing engine torque and HP primarily in the midrange. Higher lift always assists in take-off torque.

Compression Ratio vs. Cam Duration

Cylinder pressure affects power. More cylinder pressure

always increases torque and will therefore increase horsepower.

Three main factors determine cylinder pressure, which are compression ratio modified by camshaft duration and altitude. Higher altitudes lower atmospheric pressure because there is not as much air. Therefore, less air results in less compression.

As a general rule of thumb, the higher the altitude the less camshaft duration is necessary. This does not apply to visiting higher altitudes, as cam duration reduction is not possible without replacing the camshaft. Less duration is necessary when living and riding at higher altitudes most of the time.

Increasing cylinder pressure determines by more compression and less cam duration. Conversely, a decreased cylinder pressure is the result of less compression and more

cam duration.

Therefore, in performance applications more cam duration is necessary to bring in more air to burn more fuel, which creates more pressure to drive the piston down on the power stroke with increased force to turn the rear wheel with the following caveat. More cam duration will necessitate a higher compression ratio to maintain a higher cylinder pressure to maintain increased power.

## Gearing and Cam Duration

At any given speed, lower rpm riding negates the advantages of longer duration cam profiles. Lower gearing will help take advantage of duration. Higher cruising gearing will not help longer durations. Some use the analogy that tall gearing is the same as adding weight to the motorcycle because it increases the engine's workload.

Of course, there is a way to make taller gearing like that found in the supercharged gear ratios of the TC96 work for the rider. Increasing cubic inch displacement to 103c.i or bigger, with a compression increase up from stock 9.2:1 up to 10:1 or 10.5:1 with hi-lift cams relative to duration will accommodate both the taller gearing and the longer cam duration as well as those higher cfm air-flowing heads.

Stock Cam Specifications for H-D Big Twins

Stroked and/or big bore engines with more cubic inches can absorb more duration. The power band will also lower with bigger cube engines using longer duration cams. Of course, these are generalizations and we will want more compression in tandem with a longer duration cam in big-inch engines. These factors are proportional and inter-related.

* Cam timing, lift and duration specifications measure from .053” off the baseline heel of the cam lobe.

** Knight Prowler Advanced Belt Drive System cams I am currently running in my 2007 Streetglide. Check out the wild differences with these cams compared to puny stock 2007 TC96 cams.

† Panheads and 1966-1969 Shovelheads use the same cam with an identical cam lobe lift of .275" (.27466" rounded off to .275"). The difference is in rocker arm ratios. The Panhead uses a 1.500 rocker arm ratio to obtain a valve lift of .412" (.275" x 1.500 = .412"). The Shovelhead uses a 1.420 rocker arm ratio to obtain a lesser lift of .390" (.275" x 1.420 = .390"). The Knucklehead rocker arm ratio is 1.000. Therefore, cam lift and valve lift are the same.

lift)

Rocker Arm Ratio Specifications for H-D Big Twins

(Cam lift x rocker arm ratio = valve lift)

In general, we can see that much of the Evo and Twin Cam increased valve lift over previous models is due to an increased (longer), rocker arm ratio multiplication. We further increase this multiplication effect by using aftermarket greater rocker arm ratios of 1.675 and 1.750. The 1948 to 1965 Pans and Shovels had the same cam lobe lift of .275” but differing rocker arm ratios in their respective heads of 1.500 and 1.420 resulting in a Panhead valve lift of .412” and a lesser valve lift in the Shovelhead of .390”.

# Comparing Cam Timing of EFI and Carbureted Cams

Stock lift of the stock 1999 to 2006 Twin Cam cams is .473 and duration is 216-degrees on intake and 220-degrees on exhaust for both carbureted and EFI models. However, cam lobe timing specifications are different of when the intake and exhaust valves close. Overlap also changes as a result.

However, EFI cams are slightly different from carbureted ones. Their lobe profile must be exact because the delivery of fuel is precise. Carbureted cam profiles can be less exact because of the confusion reigning with inexact delivery that depends on many variables based on the physics of vacuum exploiting lower versus higher pressures to deliver fuel. Altitude and barometric pressure constantly fools with this type of fuel delivery system. There is gas and air always

hanging around in the carburetor throat and venturi. In short, there are globules of gas and air in varying mixtures, some too fat with more gas to air ratio, some too lean without enough gas relative to air ratio and some that are in correct proportions to burn creating necessary pressure to drive the piston down on the power stroke.

On demand, EFI does not encounter this inefficiency and therefore needs a more efficient cam lobe profile. Carburetors, with the properties of excess fuel just kind of hanging around allows for a less demanding cam lobe profile. As a generalization, an EFI specific cam grind will work better in a carbureted Harley than the reverse situation of a carbureted cam lobe design serving a fuel injected one.

## Comparing Cam Timing of EFI and Carbureted Cams

Stock lift of the stock 1999 to 2006 Twin Cam cams is .473 and duration is

216-degrees on intake and 220-degrees on exhaust for both carbureted and EFI models.

However, cam-lobe timing specifications are different.

## Cam Duration

Bob Wood's very successful Knight Prowler camshaft designs have lower duration figures relative to lift than most other manufacturers. Bob specializes in hi-torque camshaft design with mucho power right off the line.

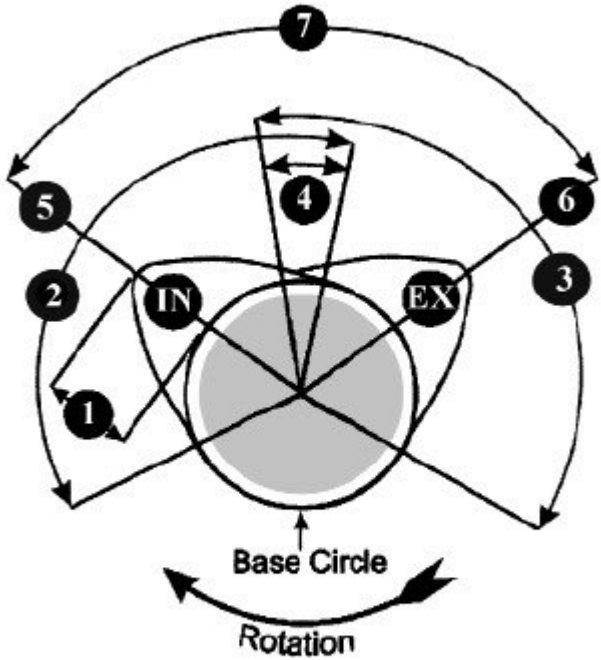
Duration is the length of time measured in degrees of

crankshaft rotation. There are two complete rotations of the crank for a total of 720-degrees to complete the four cycles of a 4-stroke engine; intake, compression, power and exhaust. Each cycle has a base of crank rotation. In both mechanical theory and strangely emissions, cam lobe profiles that approach this theoretical figure, crank rotation is a normal duration.

The theoretical duration starts and ends either when the piston in the cylinder is at TDC (top dead center) or BDC (bottom dead center). However, for reasons of mechanical efficiencies and performance objectives, cam-lobe design specifications will open the valve earlier in a cycle and close it later after the crank rotation. The degrees crank rotation a particular valve is opening before TDC in the previous cycle while its opposing valve is still open but in closing mode adds to the crank rotation to get a partial real-life duration specification since we still don’t have the total degrees yet

because I haven't accounted for the back end. Additionally, the degrees crank rotation a valve is still open on the back end after BDC in the next cycle while its opposing valve is opening adds to the above crank rotation to get a total real-life duration specification.

I will determine the intake duration for my current bike as a real-life example after we discuss overlap figures.




# CAM SPECIFICATIONS

Rotation of TC88 and TC96 camshafts is clockwise. The cam lobes extend from the base circle. The intake cam lobe (IN) is to the left of the exhaust cam lobe (EX).

1. Cam Lobe Lift is the distance measured in thousandths-of-an-inch from the base circle diameter to the highest centerline point of the elliptical shaped cam lobe. Measure the

cam lobe from the bottom of the base circle to the centerline top of the lobe and subtract the measurement of the base circle at its widest point underneath the cam lobe to obtain cam lobe lift.

2. Cam Duration of the intake cam lobe measures from where it first begins opening off the base circle until it closes onto the base circle. This measurement is in degrees of crankshaft rotation. The crankshaft needs two complete revolutions to finish all four strokes in one complete cycle of

3. Cam Duration of the exhaust cam lobe also measures in degrees of crankshaft rotation.

4. Overlap also measures in degrees of crankshaft rotation. Overlap is the number of degrees the exhaust valve is open at the end of the exhaust stroke at the same time as the intake valve is open during the beginning of its intake stroke.

5. Intake Lobe Centerline Angle (LC or LCA) measures the position of the highest or centerline point of the cam lobe in degrees of camshaft rotation. Camshaft degrees of rotation are half that of crankshaft degrees of rotation

6. Exhaust Lobe Centerline Angle (LC or LCA), positions and measures the same as the intake LCA.

7. Lobe Separation Angle (LSA), measures in camshaft degrees of rotation LSA is the distance from the intake LCA and the exhaust LCA. The Lobe Centerline Angle measures the position of the cam lobe. The Lobe Separation Angle measures the relationship between the intake and exhaust cam -lobe centerlines.

Experienced engine tuners and builders can hear the half-

cadence of camshaft rotation in relation to the full rhythm of crankshaft rotation to diagnose where a particular problem noise is emanating from.

## Valve Overlap

Overlap or more descriptively correct, valve overlap or overlap-period is the time in degrees of crankshaft rotation that the intake valve and exhaust valve are open at the same time.

Initially, one might mistakenly think there is four overlap figures in a four-stroke engine. However, the valves are theoretically closed on the upward compression stroke and the downward power stroke but with the intake valve open on the intake stroke and the exhaust valve open on the exhaust stroke. Since there are two valves in a cylinder head, one intake and one exhaust there are two overlap figures, one for

each set of valves. However, there is only one overlap figure for the Harley engine since it is the same for both heads. The opening and closing times of each valve in both heads are the same.

During the beginning of the intake stroke and at the end of the exhaust stroke both intake and exhaust valves are open. Overlap is this period-of-time in crankshaft degrees of rotation.

Overlap is Important

The valve overlap has a deep effect on how a Harley or for that matter any internal combustion engine runs. The overlap attributes will affect both high rpm and low rpm characteristics.

Let us look at the cam timing specifications for the Knight Prowler BW408G (the G

stands for gear drive and BW for Bob Wood the designer), .650” lift cams in my current bike. Cam timing for the intake is 24/44 and the exhaust is 46/22. The first figure 24 means the intake valve is open for of crank rotation before its theoretical intake cycle begins during the previous ending of the exhaust cycle also known as stroke. In this case both intake and exhaust valves are open for of crank rotation. The second figure 44 means that the intake valve stays open for of crank rotation after its theoretical intake cycle ends during the next cycle called the compression stroke.

Duration is now easy to figure out for the intake cycle since we add of crank rotation to the duration of the intake cycle and include the of crank rotation for a total of real-life duration.

The exhaust timing figures of 46/22 indicate that the exhaust valve is open for of crank rotation before its theoretical exhaust cycle begins during the previous ending of the power cycle. The second figure 22 means that the exhaust valve stays open for of crank rotation after its theoretical exhaust cycle ends during the next cycle of the intake stroke. In this case, both the intake and exhaust valves are open at the same time for of crank rotation

We add the intake valve opening of of crank rotation before TDC to the exhaust valve's closing of of crank rotation after its theoretical exhaust cycle ends after TDC to obtain the valve overlap of of crank rotation for this particular cam.

Many ignore valve overlap, but it is critical to making power. A good flowing intake tract will not need as much

overlap, while a restricted poorly flowing intake tract would suffer from a cam with too much overlap.

When observing the valve train events at their most basic levels, the closing of the intake valve has the most effect on power. Closing the intake too late will result in power loss because the piston will force the incoming fresh intake charge right back out through the carb or EFI throttle body from an effect known as Closing the intake valve too soon result in the cylinders inability to breathe in all that fresh air and fuel.

A perfect engine will shut the intake valve precisely as the air/fuel intake stops filling the cylinder. However, finding that point is difficult, to say the least because the entire intake tract, from the air cleaner into the cylinder head, and even the exhaust system all have an effect on when the intake valve should close.

The closing time will vary from engine to engine depending on the combination of performance parts utilized.

The Knight Prowler Advanced Belt Drive System cams I am currently using in my bike.

Bob Wood designs these hot-rod-cams for 96-124 cubic inches with appropriate gearing and minimum 10.5:1 compression. The overlap figures are timing specifications in degrees of crankshaft rotation measured at the traditional .053” off the baseline of the cam. Lift and duration figures are usually measured beginning at .053” in the Harley world also. Purists feel that the affect off this will compromise true overlap figures and analysis. To complicate this a little further, many performance engine builders gauge a race cam’s

profile aggressiveness by looking at cam duration at .020”, .053” and .200”. The .020” number determines how responsive the motor will be and how much low end torque the motor will make. The .053” number estimates where peak power will occur, and the .200” number gives an estimate of the power potential. A TW-408G has the same exact specifications as a BW-408B. The first set of cams are for a traditional gear drive while the second set are for the Advanced Belt Drive System where the rear cam is driven by the belt drive on the outboard cam support plate while the inner connecting drive between rear and the front cam is via gear drive.

Calculating Valve or Cam Lobe Overlap

There are different formulae to calculate the overlap of the cam also known as valve overlap. The formula depends on the information available. Purists do not use figures taken beginning at .053”. They feel that true overlap figures require

starting at on the base circle seat of the cam lobe not at .053” after it. I mention this to be correct but we will use the .053” opening specification since it is real world. It is the way that the H-D world does it. It would be confusing for our purposes to do it any other way.

Use the first formula if knowing the opening and closing times of the valves.

For example, on my current BW-408B cams, the cam timing specifications are 24/44 for the intake valve and 46/22 for the exhaust valve. These figures mean that the intake valve opens (crankshaft rotation), BTDC (before top dead center), on the previous exhaust stroke as it is completing its cycle. The intake valve closes ABDC (after bottom dead center), of the intake stroke while the compression stroke has begun its journey upwards. The exhaust valve opens BBDC (before bottom dead center), on the previous power stroke, as it is finishing. This valve closes ATDC as the next cycle, which

is when the intake stroke is beginning.

If we know the intake, opening time and the exhaust closing time, the formula is:

Calculate Valve Overlap

Using cam timing specifications

(Example; cam timing intake 24/44 and exhaust 46/22)

Intake opening time + exhaust closing time = valve overlap

IO + EC = VO

+ = valve overlap

I am going to jump ahead a bit here into lobe centerlines and lobe separation angles to demonstrate another way to calculate valve overlap.

These are simple concepts.

The lobe centerline or LC is the angle of the highest point of the cam lobe.

Lobe Separation Angle or LSA is the distance between the lobe centerline of the intake cam lobe and the centerline of the exhaust cam lobe.

If we know the intake and exhaust duration and the Lobe Separation Angle (LSA), we can once again calculate valve overlap. Not all cam manufacturers list the LSA.

I will demonstrate how to calculate LSA in that section further on in this chapter.

At the risk of confusing the valve overlap calculations, I will show briefly how LSA is calculated.

However, it gets deeper before its necessary time as we need to add the intake and exhaust Lobe Centerline figures, then divide their sum by 2 to determine the LSA.

Calculate Lobe Separation Angle (LSA)

(Using BW408 cams specifications listed in above chart)

Exhaust centerline + Intake centerline ÷ 2 = lobe separation

angle

+ ÷ 2 =

Now that we know the LSA, we can also calculate valve overlap another way.

Calculate Valve Overlap

Using cam duration & Lobe Separation Angle specifications

(Using BW408 cams specifications listed in above chart)

Intake duration + exhaust duration ÷ 4 – Lobe Separation Angle x 2 = valve overlap

I.D. + E.D. ÷ 4 – LSA x 2 = VO

$$+ \div 4 = -101 \times 2 = \text{Valve Overlap}$$

Well, I think we have had enough of mathematics for a while. I am going to back up a little to describe the actions and functions of the valve train in order to explain some background information about how the dictates of the elliptical-shaped cam lobes transfer to their respective valves.

The Valve Train

A tappet roller rolls over the egg-shaped cam lobe. The tappet roller pivots on an axle on the lowest part of the solid or hydraulic lifter also known as a The hydraulic lifter actuates hydraulically with engine oil in order to adjust the valve train. The hydraulic action reduces lash or spacing between mating parts at a minimum for both purposes of

engine efficiency and noise reduction. Constant adjustment results from engine heating expansion and cooling contraction of interconnecting parts in the valve train.

The pushrod's lower balled end sits in a mating seat on the top portion of the hydraulic lifter. It extends from the timing chest up into the head encased in an oiled and dirt free chrome pushrod-tube housing. Stock Twin Cam pushrods are of fixed length while aftermarket performance pushrods are adjustable for ease of installation. Secondly, they also accommodate higher lift performance camshaft lobes.

Interestingly and for what reason I cannot fathom, except as a cost saving measure, the H-D CVO 110" engines use stock Twin Cam fixed length pushrods. I discuss this at length in Chapter H-D 110 Cubic Inch CVO, subsection, Disassembling the CVO 110: What I Found.

The upper-balled end of the pushrod fits into a mating pocket in one end of the rocker arm, which swivels up and down on a rocker arm shaft a distance equal to the cam lobe lift. One main mechanical reason for a rocker arm is it connects the pushrod on the opposite side of the head to where the valve is locates on the other side. The approximately 3 ⅝-inch long rocker arm transfers the dictates of the rotating cam lobe to the top portion of the valve stem encased in the single beehive valve spring, 2004 and later or dual cylindrical springs, 2003 and earlier.

The stock rocker arm end-pad opposite to the pushrod side sits over the valve stem. The pad rubs along and down on the top of the angled valve stem, forcing it to open.

The function of the valve spring(s) is to close the valve back onto its valve seat after the elliptical cam lobe rotates past its highest point, compressing the springs. The springs push the valve closed as the valve train follows the descending cam lobe down.

If the rocker arm ratio were 1:1; that is, the extension of the rocker arm that cups over the upper balled-pushrod was on the same plane as the rocker arm extension on the other side that actuates the valve, then cam lift and valve lift would be equal or the same figure. This is the case with the 1936 to 1947 Knuckleheads. However, on every Big Twin from the 1948 Panhead to present, one side of the rocker arm offsets to the other so a multiplication of the cam lift transfers to the

valve stem thus increasing valve-opening lift.

The stock Twin Cam's rocker arm ratio is 1.625:1. Therefore, the cam lift multiplies by 1.625 to achieve the effective valve lift or valve opening.

Performance rocker arms have roller ends instead of a metal rubbing pad to actuate the valve stems. They may also have altered rocker arm ratios such as an Ultima roller rocker arm with an increase of .050" ratio to 1.675 rocker arm ratios. Another example is a Crane roller rocker arm with a ratio of 1.750, an increase of .125" ratio over stock. I use increased rocker ratio arms effectively to enhance lift when unsatisfied with too long, for my purposes, duration figures of a particular cam.

Enhancing a stock cam's valve lift by using increased ratio

rocker arms is one way of lowering the power band into more useable territory and gaining extra low-end torque. Therefore, I use increases in rocker arm ratios to boost torque and lower the power band of a particular camshaft.

It is important that the valve springs can handle the increase in valve lift without the springs coil binding or the upper valve collar hitting the top of the valve guide or the valve extending too far into the combustion chamber and hitting its mating piston valve pocket.

As I have said before, I feel many production cams favour touring over city-street riding. They do this by having longer duration figures relative to lift. In truth, I guess it would be fair to say they are compromise camshafts.

Performance is not about compromise. Actually, the last statement is not a fair one either since compromise is part of

any decision-making process to coordinate parts for the best and safe performance gains. To clarify, once the performance mechanic has decided on the type of riding the engine design is for, he or she will choose a specific cam for that purpose not a compromise

## Lobe Centerline Angle (LCA)

The Lobe Centerline Angle determines two important valve-timing characteristics. The LCA dictates the valve overlap around TDC. Secondly, LCA establishes the amount of intake or exhaust valve closure delay there is beyond the theoretical end of the applicable stroke.

Hi-torque cams have lobe centerlines around For example, my current Knight Prowler TW408G cams have an intake centerline and exhaust LC of

Lobe Centerlines (LC) also known as Lobe Centerline Angle (LCA) is not the same as a Lobe Separation Angle (LSA). These two terms often confuse. However, LC and LSA directly connect.

The two have very similar names but they are very different. LC and LSA control different events in the engine. Lobe Separation Angle is simply what it says. LSA is the numbers of camshaft degrees separating the peak lift points of the exhaust lobe and the peak point of the intake lobe. The peak points are also known as the lobe centers of the cam intake and exhaust lobes. The distance between the two is the Lobe Separation Angle.

Changing the LSA of a cam is impossible, as it is set when the cam is ground. LSA cannot change by the mechanic degreeing the cam in the engine.

The intake centerline, on the other hand, is the position of the centerline, or peak lift point, of the intake lobe in relation to top dead center of the piston. Degreeing the intake centerline of the cam into the engine forward or back of the stock centerline position will change timing to the specific needs of the engine function. The highest tip on the elliptically shaped cam lobe also known, as the point of maximum lift is the lobe centerline.

Therefore, on a Twin Cam, a two-cylinder engine with four valves actuated by four mating cam lobes there are four lobe centerlines or LC's. In a Harley-Davidson the front and rear cylinders, use the same cam lobe centerlines so there is one intake cam Lobe Centerline Angle figure and one exhaust cam LCA.

In review and for comparison, the Lobe Separation Angle or LSA measures in degrees of camshaft rotation between the exhaust lobe centerline and the intake lobe centerline.

Remember that camshaft rotation bases on to complete a full cycle while crankshaft rotation bases on two revolutions of the engine or rotations to complete its cycle.

Only a few minutes ago, these two definitions seemed overwhelming to the novice and now the simplicity of LC and LSA meanings are readily apparent.

So, how do we express and figure out lobe centerline angles? Well, now life gets interesting and I get to show off just how easily Bob Wood simplifies calculations. I will use Bob's bang-on simplistic calculation methodology and frame it beside the ways the mathematicians use.

Please learn one thing from this in advance. The Wood Performance way does not require a calculator to do LC's and LSA's. Bob un-confused me in an email in late 2007 when I queried him about the LCA and LSA figures for the BW-408B Knight Prowler Advanced Belt Drive System cams in my bike.

Bob succinctly wrote that the "Intake LCA is 100 degrees ATDC and exhaust is 102 degrees making the LSA 101 degrees for the BW-408B.

This is extremely simple. This will make you a cam-timing wizard!!!!!!!!!!!

A: Simply look at the "INTAKE DURATION " which is 248 deg. DIVIDE BY 2 = 124 degrees. (Donny) DO IT IN YOUR HEAD!!!!!

B: SUBTRACT INTAKE OPENING OF 24 degrees.

ANSWER: 100 degrees INTAKE LCA.

C: Now take “EXHAUST DURATION” WHICH IS 248 degrees also, DIVIDE BY 2 = 124 degrees. “(Donny) DO IT IN YOUR HEAD!!!!!!

D: SUBTRACT EXHAUST “CLOSING” of 22 degrees. (The) ANSWER IS 102 degrees EXHAUST LCA.” 2

Now we are able to do the LSA calculation once we know the lobe centerline angles.

Bob continues “E: INTAKE LCA of 100 degrees & EXHAUST LCA of 102 degrees = 100 deg + 102 degrees = 202 degrees DIVIDED BY 2 = 101 LSA.

(Donny)...NOW YOUSE IS A CAM

WIZZZZARD!!!!!!!!!!!!!!!!!!!!!!” 3

Bob Wood is a pleasure to talk to as he reduces normally complicated stuff to simple basics.

Example; the cam specifications are for the BW-408B Knight Prowler Advanced Belt Drive System.

Calculate Lobe Centerline Angle (LCA)

(Example calculation uses BW-408B cam specifications)

Intake LCA

Intake duration ÷ 2 – intake opening = Intake LCA

$\div\ 2 = - =$ Intake LCA

Exhaust LCA

Exhaust duration $\div$ 2 – exhaust closing $=$ Exhaust LCA

$\div\ 2 = - =$ Exhaust LCA

Another similarity of measurement and expression of LC and LSA is that both do this in degrees of camshaft rotation.

Generally, LSA's are between and

Another major difference between an LC and an LSA is that a mechanic can adjust lobe centerlines in tandem to advance or retard cam timing by degreeing the camshaft. I

will explore degreeing camshafts a little further on.

However, an LSA is non-adjustable because it is ground into the cam lobe profiles fixing the intake lobe and exhaust cam lobe in place. I will expound on this subject a little later on also.

Valve Closing Delay, Overlap, and LCA

As I said earlier and it bears repeating that, the Lobe Centerline Angle affects valve overlap around TDC.

In addition, the LCA establishes the amount of intake or exhaust valve closure delay there is beyond the theoretical end of the relevant stroke.

As the piston descends to the bottom of the cylinder but before the piston reaches the bottom, the exhaust valve begins to open. Most of the fuel has burned. Cylinder pressure will begin to push these burnt exhaust gases out into the exhaust port and the exhaust headers into the mufflers. After the piston passes Bottom Dead Center (BDC), it begins ascending the cylinder on the exhaust stroke. This upwards action forces the remainder of the spent exhaust gases out of the cylinder and combustion chamber to make room for the fresh, clean charge of air and gas.

As the piston moves toward the top of the cylinder, the already opening exhaust valve goes through maximum lift and begins to close. Just before the piston reaches the top of its exhaust stroke (TDC), the intake valve begins to open but the exhaust valve is not yet fully closed.

Theoretically, this should not work unless other forces of physics assist like the Principle of The piston’s exhaust stroke

is removing just about all of the spent charge by the time the piston approaches top-dead-center. However, just before this, the intake valve begins to open slowly. Scavenging begins. The remaining exhaust gases rushing out into the exhaust will draw or suck in the start of the intake charge.

Conversely, the developing momentum of the intake charge also assists in pushing out the remaining exhaust gases. It becomes a symbiotic affair, first with exhaust pulling and then a little intake pushing.

Scavenging is how the engine flushes out all of the spent fuel charge. Again, some of the new intake charge gases may escape into the exhaust. Thus, valve opening and closing timing is crucial.

Finding the sweet spot where pollutants fully exit but the

beginning of the fresh intake charge is not lost is the science and many times the art of valve timing.

Once the piston passes through the exhaust stroke ending theoretically at TDC and starts back down on the intake stroke, the intake charge pulls in quickly. Therefore, the exhaust valve must close at precisely the right point after the TDC to ensure prevention of any burnt gas re-entering.

Overlap is this area around TDC where both intake and exhaust valves are open.

This is one of the most critical moments in the engine's rotational cycle. Intake and exhaust valve opening and closing must position correctly with the top-dead-center position of the piston. To complicate matters the function design of the motorcycle changes this positioning for different types of riding and other factors.

As the engine passes through the overlap phase, the intake valve is opening very quickly while the exhaust valve closes just after the piston starts descending.

As the reader knows, the intake stroke is where the engine breathes in yet another fresh charge of virgin air/gas mixture. The intake valve reaches its maximum lift at some planned point, which is in and around 105-degrees of crankshaft rotation after top dead center. This degree-measurement is the intake centerline lobe also known as the lobe centerline (LC) or Lobe Centerline Angle (LCA). This refers to the position of cam installation also known as degreeing in the engine in relation to crankshaft rotation.

All moving valve-train engine components time off the positioning of the crankshaft.

LCA determines valve overlap around piston TDC and has much influence on the amount of intake or exhaust valve closure delay there is beyond the theoretical end of the relevant stroke. Furthermore, tighter or closer LCAs produce earlier intake valve closure after BDC. A hi-torque low rpm cam needs a tighter LCA's. Wider or further apart Lobe Centerline Angles extend the intake valve closing beyond BDC. This will move the peak torque of the power band up the rpm range.

Cylinder Pressure and Intake Valve Closing

The piston descends all the way to the BDC.

As the piston begins its ascension in the cylinder bore on its compression stroke, the intake valve is fast returning towards its mating valve seat. The closing point of the intake

valve will determine where the cylinder actually begins to build pressure.

After the intake charge is in the cylinder, the intake and exhaust valves close. However, it is very important to have the intake valve open long enough to fill the chamber. It is equally important that it close it soon enough to yield maximum cylinder pressure. This is a very tricky valve-timing point in the camshaft cycle.

The power builds as the ascending piston begins to compress the mixture. Compression, where the piston begins to build pressure is crucial to power.

Before TDC on the compression stroke, ignition sparks the plug. How far before TDC ignition occurs is ignition timing. Ignition timing will change frequently during the ignition

curve. This advance or retard ignition timing is necessary because of many factors needing more or less time for the flame travel to move across the combustion chamber. Acceleration will move the piston faster so spark must occur sooner also. Conversely, decelerating allows the spark to ignite later.

A big bore kit whether it is the TC88 to TC95 or the TC96 to TC103 conversions will increase the diameter of the cylinder to create more swept volume or displacement. The flame travel needs more time to travel the extra distance so the spark must ignite the fuel mixture sooner.

These few examples are some of many.

All the various factors affecting ignition timing will end in compromise spark timing. Decelerating rpms and thus later spark ignition must compromise with the earlier spark timing

necessary for big bore and on it goes through a myriad of demands.

Production Cams–Compromise Cams

Many times, I discuss my dissatisfaction with many cam manufacturers and the grinds they produce that just are not quite, what I am looking for.

Power bands rise away from low-end power and torque. The cams I refer to are wonderful in high rpm, fast cruising situations. For me, I would like to lower the power band a little to retain the higher speed aspects with some off the line oomph.

Bob Wood with his series of Knight Prowler cams is an exception. He concentrates on lowering the power band with

lots of low-end torque. Harley-Davidson's are heavy machines and many of the riders are far from lightweights. I want power to get me going in lower rpm ranges.

Marketing sensitivity on the part of cam manufacturers quite often lead to compromise cam profiles with wide lobe centerline angles. This is not necessarily a bad thing since most do not really know what they are after in matching a cam to their engine. However, for those in the know it can be quite annoying. Therefore, I feel some companies look to balance horsepower and torque with of all things, fuel economy and idle quality while at the same time assuming we all cruise at higher speeds.

Very often, the choice of wide LCA's comes with sacrificing some of the potential power increase for idle quality. They will do this by grinding their performance street profiles on wide LCA's. They sacrifice some of the potential power increase in favor of a very smooth idle.

Me, I like a little rumbling roughness at idle in favor of more power. My Harley's exude toughness as they growl at idle, shuddering with pulsating waves, just waiting to pound-the-pavement. However, the grind profile still allows idling without stalling while giving low-end torque and only minimally compromises high-end rpm. In short, the power band is exactly where I want it in everyday usable territory.

Therefore, wide LCA's are not the only way to go.

Not everyone, like me, wants the smoothest idle and the highest intake manifold vacuum possible. I personally want tighter LCA's for my style of aggressive riding to increase my low-end and maximum torque with more cranking compression.

If you are a Bagger rider and have cammed your bike, you may have noticed that most cam manufacturers offer a hi-torque mild bolt-in camshaft for heavy bikes. For this purpose, they grind these cam profiles with less duration and with a tighter LCA's to produce more low-end torque.

A tighter LCA is suitable if the LCA overlap improvement and the duration combination is not extreme. The tighter the LCA chosen, the shorter the cam duration must be to preserve idle. This has the effect of developing close to the same overlap given by a longer duration and a wider LCA.

Again, the trade-off low-end torque gain will be proportional to the loss of some top end horsepower. This will be more evident on pre-2006 Dynas and pre-2007 Twin Cams because of their gear ratios.

The 2006 Dynas and all 2007 later Twin Cams have supercharged gear ratios that accommodate these tighter LCA cams wonderfully.

The Wood Knight Prowler BW-408B cams in my current bike have a wild .650" lift up from the stock TC96 .471"/.474" intake/exhaust lifts, rather mild duration compared with other manufactures that produce big-lift cams with a very tight intake/exhaust LCA's. My Streetglide is fast off the line and is faster than my previous Baggers approaching and going beyond 100 mph. I attribute the speed in combination with other changes to Bob Wood's camshafts combining with the TC96 gear ratios and a few other ditties.

Please see Volume I: The Twin Cam, Chapter VII: The Controversial 6-speed Transmission and in this Volume II: Performancing the Twin Cam, Chapter X: The Power subsection, Gear Ratios for more

In conclusion, from a broad perspective, lobe centerline angles strongly tie in with the flow capabilities of the cylinder heads relative to the cylinder swept volume needs. Many refer to this as the ratio of cubic-feet-per-minute (cfm) or air-per-cubic-inch displacement, (cfm per c.i).

## Degreeing Cams

The mechanic installing the cams can move or adjust lobe centerlines around by rotating the cam gear in either direction a few degrees. The positioning of this gear attached to the camshaft if pressed on will affect cam timing. Some cams, notably 1999-2006 Twin Cam cams excepting the 2006 Dyna series have their cam sprockets cast onto the camshafts so they are non-adjustable...well, sort of.

Cam timing can adjust via an advanced rear cam-drive gear. Although, the 2006 and all 2007 to present Twin Cam cams have pressed on cam gears, it is much easier to change the rear-cam drive gear timing when using stock Factory camshafts.

Another way is to position the cam forward or backward of its timing mark indent with its mating drive sprocket. We accomplish adjusting cam lobe centerlines by advancing or retarding cam timing. First, the mechanic uses a degree wheel to ensure the cam installs at the manufacturer's suggested intake cam-lobe centerline, which is the maximum point of lift of the intake lobe.

For example, if the cam manufacturer suggests the cam intake cam-lobe centerline is the degree wheel will ensure the intake cam lobe's highest point installs at 100-degrees. This is the stock setting.

The mechanic can advance cam timing by lowering the install point a few degrees below or can retard the cam timing by increasing the install point a few degrees above 100-degrees; how many degrees will become the question. There is usually up to four degrees practical variation either way. Advancing or retarding cam timing will shift the power band up or down for the particular needs of a bike or riding style.

Although I said earlier that an LSA is non-adjustable because it is ground into the cam lobe profiles fixing the intake lobe and exhaust cam lobe in place, nevertheless there is a way to do adjustments at the manufacturing level. The manufacturer may grind advance timing or retard timing into the intake cam lobe as part of the cam profile to raise or lower the power band a little. The mechanic cannot reverse what the manufacturer does with LSA's whether stock, advanced or

retarded.

However, we can adjust Lobe Centerline Angles equally by up to 4-degrees either way by advancing or retarding cam gear timing during installation

As usual, the aftermarket comes up with innovative performance options. Bob Wood's Knight Prowler optional Gear Drive setups are available with and advance and retard keys.

- The Wood Performance, 2006 Dyna and all 2007 to present Twin Cam, 216020 + 4 cam gear advances both cams 4-crankshaft rotational degrees.

- Part number 288015 + 4, fits all Twin Cams from 1999 to 2006 except the 2006 Dyna. Either cam gear fits on any stock or performanced camshaft.

The timing advance gives both higher cranking compression and running compression resulting in more torque and horsepower in the lower rpm range.

## Calculation of Intake Cam Timing Change

Calculate intake lobe advance or retard timing during manufacturing.

Assume there are an LSA and an intake Lobe Centerline of 112 degrees.

Cam Lobe Separation Angle - Intake Cam Lobe Centerline = Advance/retard Cam Timing

LSA – ILC = Timing change

– = retard timing

Assume there are an LSA and an intake Lobe Centerline of 110 degrees.

LSA – ILC = Timing change

– = advance timing

Once again, let us look at the BW-408B Knight Prowler Advanced Belt Drive System cam specifications, which have an LSA of and an intake LC of 100-degrees.

LSA – ILC = Timing change

- = advance timing BW-408B cams

In conclusion, LSA changes are not possible unless it is ground into the intake cam-lobe profile during the manufacturing process.

However, the mechanic can advance or retard the cam timing in relation to the crank when degreeing in the lobe centerline during cam installation.

## Advancing or Retarding Cam Timing Consequences

Advancing the cam to close the intake valve sooner, improves low rpm performance (torque), by reducing cylinder pressure losses. This improves the corrected compression

ratio, which suffers by cylinder volume loss through open valves. Retarding cam-timing delays the intake valve closure and improves high rpm performance. Scavenging, sometimes called ram effect, increases cylinder fill at higher rpms and with the intake valve opening longer. The intake charge air/gas stacks up in the cylinder as fast exiting exhaust gases draw intake fuel and air into the cylinder.

Comparing Cam Construction

The pictured upper cam is a stock 1999-2006 except 2006 Dyna uni-construction cam. The complete unit is cast as one and then CNC machined and hardened. The drawback with this design is that there are differing hardening requirements for the rotating cam lobes, the cam chain sprocket and the end cam journal bearing surfaces. In practice, this design worked okay except the loose fit of the sprocket in the silent timing chains allows up to a variance in cam timing wander. The bottom 2006 Dyna and all 2007 to present TC96 cam comprises of five separate pieces. Hardening of cam lobes, sprocket, and end journals is now specific to the needs of the component. The sprocket is thinner than its predecessor sprocket and fits better with its mating roller chain but there is still wandering cam-timing variance. Notice the right side cam outboard journal ends of both cams. The early cam has a smaller end journal as it fits into an actual roller or ball bearing that house in the cam support plate. The later cam has a much larger cam journal that sits in its cam support plate parent material Parent material is what it says; there is

no actual bearing but a hole in the support plate. There is much more on this subject in Chapter II: Repairing the Achilles Heel, subsection, Moment Arm Torsional Bearing The inboard cam journals also have different sizing. The early cam journal is .875” in diameter while the late one increases to a full 1.000”. The end journal rotates in INA SCE148 and SCE168 early and late bearings in the rear of the timing chest of the right side engine casing. I always replace these bearings with, in my opinion, superior Torrington B148 and B168 bearings respectively. In between the upper and lower cams is a mostly disassembled late cam. The cam lobes and sprocket will press onto the mainshaft in the same places as the complete cams. Timing these separate lobes to the shaft is very precise work. The inboard journal end is also the fifth separate piece but presses on so tightly, I could not remove it without wrecking it and the shaft. The visible slot on the separated shaft allows a puller to grab on for removal.

Engine Displacement and Compression Ratios

Compression ratios affect cam choice, especially with duration specifications, more than the size of the engine although displacement is also a factor.

Higher compression and long durations work well together. Longer duration cams do not do well with lower compression. This is why many cam manufacturers' will specify compression ratio ranges suitable for their longer duration cam profiles.

Conversely, longer duration cams will allow easier starting of high compression engines because these cams hold the valves open longer allowing some volume in the cylinders to escape. Thus, the real life corrected compression ratios are lower than the static compression ratios, which only compare the volumes of the cylinder and combustion chamber with the combustion chamber volume.

Static Compression ratio = cylinder volume + combustion chamber volume ÷ combustion chamber volume

Compression has many names and meanings. There are many ways of interpreting compression measurements. There is even more terminology describing the principle. The static compression ratio compares the volume of the cylinder and combustion chamber to the volume of the combustion chamber alone.

Static Compression Ratio

Suppose we have an 840cc cylinder and an 85cc combustion chamber. To calculate the static compression ratio we add 840 + 85 = 925. We then divide 925 by 85 = 10.9:1 to obtain the static compression ratio also known as,

- Theoretical compression ratio or
- Mechanical compression ratio as well as
- Advertised compression ratio or
- Nominal compression

Corrected Compression Ratio

The corrected or effective compression ratio is more accurate for a real-life engine. It recognizes that cylinder volume is lost with open valves. The math to calculate this is complicated requiring a computer program. JIMS (www.jimsusa.com), as well as other motorcycle distributors, has a computer program for this and many other calculations

known as Accelerator.

Cranking Compression Ratio

Then we have cranking which measures in psi (pounds per square inch), actual compression with a tool called a compression gauge. The technician screws in or presses a hose connecting to a gauge into the spark plug hole and cranks the engine over to get a dry cranking compression in a psi reading.

Wet cranking compression is usually higher than dry testing because the technician squirts oil into the cylinder beforehand so the rings can seal better.

Running Compression Ratio

Running compression is what it says. Running compression is actual real-time compression under riding conditions where there is heat expansion and ring sealing against cylinder walls with oil lubrication.

## Altitude Effects on Various Compression Ratios

Altitude lowers corrected compression ratios because there is progressively less air with rising elevation. Altitude will not affect static compression ratios, as it is just a comparison of volumes in the cylinder and combustion chamber relative to each other. To counter the loss of effective compression from longer cam durations and overlap, thinking mechanics will increase static compression ratios.

## Generalized Cam Profile Effects on Engine Operation

With certain traits, a positive effect may become a negative one or vice versa depending on the type of engine in use.

## The Power Band

*A tight LCA increases torque with a rougher idle and the danger of exhaust reversion back into the chamber bastardizing the fresh intake charge. A wider Lobe Centerline Angle causes the intake valve to stay open longer after-

bottom-dead-center (ABDC), of the intake stroke. A stock engine receives up to an approximate 10% fill after intake BDC while performanced engines receive more fill, possibly up to and past 20% extra fill. Greater increases are negated by the now rising piston on its compression stroke. Too wide an LCA produces the possibility of intake reversion where the intake air/fuel charge may force back up and out of the intake tract reducing potential power, specifically

**Compression ratio is a catchall phrase. There are a whole bunch of ways of looking at compression measurements and even more terminology referring to all this stuff. In order to understand what corrected compression ratio refers to; I will look at static compression ratio first. The static compression ratio compares the volume of the cylinder and combustion chamber to the volume of the combustion chamber alone. Suppose we have an 840cc cylinder and an 85cc combustion chamber. To calculate the static compression ratio we add 840

+ 85 = 925. We then divide 925 by 85 = 10.9:1 to obtain the static compression ratio also known as theoretical or mechanical compression ratio as well as advertised or nominal compression The corrected or effective compression ratio is more accurate for a real-life engine. It recognizes that cylinder volume is lost with open valves. The math to calculate this is complicated requiring a computer program. Then we have cranking which measures in psi (pounds per square inch), actual compression with a tool called a compression gauge. The technician screws in or presses a hose connecting to a gauge into the spark plug hole and cranks the engine over to get a dry cranking compression in a psi reading. Wet cranking compression is usually higher than dry testing because the technician squirts oil into the cylinder beforehand so the rings can seal better. Running compression is what it says. Running compression is actual real-time compression under riding conditions where there is heat expansion and ring sealing against cylinder walls with oil lubrication. Altitude lowers corrected compression ratios because there is progressively

less air with rising elevation. Altitude will not affect static compression ratios, as it is just a comparison of volumes in the cylinder and combustion chamber relative to each other. To counter the loss of effective compression from longer cam durations and overlap, thinking mechanics will increase static compression ratios.

## Cam Lobe Separation Angle (LSA)

The acronym LSA refers to Lobe Separation Angle also known as the Lobe Displacement Angle or the Lobe I try to include the various vernaculars referring to the same principle because nothing is as confusing as interchanging these different terms if not realizing that they are actually synonyms meaning the same thing.

A tight LSA has the respective intake and exhaust cam

lobes (centerlines), closer together.

A wide LSA has a wider angle between the high points (LC's), of the respective cam lobes.

Other factors equal, tightening the LSA produces more valve overlap around TDC, while widening the LSA results in less overlap.

At bottom dead center of the intake cycle, a wide LSA produces a longer delay to valve closure after the piston has passed BDC

The calculation of LSA gets deeper before its necessary time as we need to add the intake and exhaust Lobe Centerline figures, then divide their sum by 2 to determine the LSA. We learned a little earlier that to calculate the Lobe Centerline Angle for the intake, we divide the intake duration

by two and then subtract the intake valve opening timing to determine intake lobe centerline, which is the maximum lift-point of the intake cam lobe.

Determine the exhaust LCA by dividing the exhaust duration by two and subtract the closing timing to get the exhaust Lobe Centerline Angle, which is the maximum lift-point of the exhaust cam lobe.

Calculate LSA

(Example calculation uses BW-408B cam specifications)

Intake LCA + Exhaust LCA ÷ 2 = LSA

- 2 = LSA

Overlap Controls Engine Vacuum

Engine cylinder vacuum, which is a necessary function for cylinder fill, is directly controlled by overlap.

However, Lobe Separation Angles (LSA) is not the same as overlap.

Ably assisting engine vacuum that initially brings in the intake charge is the scavenging effect, a colloquial mechanical term, more properly known as the physics Principle of Momentum. Scavenging creates the high velocity momentum of exhaust gases rushing out the exhaust port to create more vacuum suction behind the exiting exhaust gases that pulls the fresh air/gas charge into the chamber at higher rpms.

This is very important since scavenging directly relates to

the cylinder heads, exhaust system, and engine breathing capabilities. If there are weak cylinder heads in terms of flow like those found in the TC88 and to a lesser extent in the TC96 and an improper poor scavenging exhaust system, the most obvious example being drag pipes on a street bike, the wrong LSA might not put enough fresh charge into the cylinders.

Typically, a wider Lobe Separation Angle of to 116-degrees will mean less overlap and higher vacuum at idle. A narrower lobe separation to provides just the opposite.

Wild full-race cams may be the exception. They can have a wide Lobe Separation Angle like, say 118-degrees and still have lots of overlap and make little vacuum at idle because the intake valve has much duration, opening very early in the cycle and not closing until long past when the exhaust valve

begins to open. These engines have a particular rumbling sound at idle that we all love because of the intake and exhaust tracts working together. In old Harley vernacular, we refer to this as a lumpy cam.

Lobe Centerlines (LC) or Lobe Centerline Angle's (LCA) are the maximum lift high points of the intake and exhaust cam lobes. I am repeating definitions on purpose to keep the reader and myself sometimes oriented, as this stuff can be tough until acclimated. With street cams, the wider you move the lobe centers apart, increasing the Lobe Separation Angle such as to 116-degrees, the less overlap you typically have.

Although both wide to and narrow to lobe separation angles do have an effect on overlap; it is still possible to get overlap with very wide Lobe Separation Angle race cams. These LSA figures for wide and narrow angles are approximate and used for comparative purposes. Some LSA's are narrower than 101- degrees of crankshaft rotation or

wider than 116 degrees.

The overlap phase begins with the opening intake valve. The timing of this is very critical to vacuum, throttle response, emissions and particularly, gas mileage.

The area between the intake opening and the exhaust closing, and where it occurs, which is the amount of overlap is one of the most significant markers in the engine cycle. If the intake valve opens, too early, exiting exhaust gases will push the new intake air/gas charge into the intake manifold. If the intake valve opens too late, the cylinder will lean thus negatively hampering engine performance.

If the exhaust valve closes too early it will trap some of the spent gases in the combustion chamber bastardizing the new intake charge. If the exhaust valve closes too late it will over-

scavenge the chamber; taking out some of the fresh intake charge, again creating an artificially lean condition.

If the overlap phase occurs too early, an overly rich condition will develop in the exhaust port, negatively affecting gas mileage, which will be a big no-no to emissions controls.

Therefore, it becomes easy to understand that everything about overlap is critical to the performance of the engine.

Hi-torque cams never have more than a separation between the exhaust Cam Lobe Centerline and the intake Cam Lobe Centerline.

## Calculating Centerline Separation

(Example calculation uses BW-408B cam specifications)

Exhaust centerline – Intake centerline = separation

=

Calculating Lobe Separation Angle (LSA)

(Example calculation uses BW-408B cam specifications)

Intake Lobe Centerline + exhaust Lobe Centerline ÷ 2 =

Lobe Separation Angle

intake Centerline + exhaust Centerline = ÷ 2 = LSA

Reviewing Terminology

- We measure LSA in camshaft gear rotational degrees of for a complete cycle not to confuse with other engine cycle measurements that measure in crank rotational degrees of to complete a full

- The crankshaft or flywheel assembly moves at twice the rotational speed that the camshaft will rotate. If I say this in the opposing way, the camshaft rotates at half time of the crank rotation.

The experienced mechanic will know this attribute of rotational speed and can listen to the cadence of running engine components and easier isolate symptomatic noise to ensure a correct diagnosis. To the uninitiated, it is just all noise.

- As I have said, the Lobe Separation Angle is the distance

in camshaft degrees between the highest point of lift on the intake cam lobe and the maximum lift on the exhaust lobe.

- LC or lobe centerlines only describe the position of each cam lobe.

- LSA or Lobe Separation Angle describes the relationship between the intake and exhaust lobe centerlines.

- Lift and duration only describe the characteristics of an individual cam lobe while overlap describes how long the intake and exhaust valves are open at the same time.

To understand this better, an LSA of will mean the intake and exhaust lobes align perfectly opening and closing their respective valves identically at the same time. In this case, each LC will have the same lobe centerline angle. Of course,

this is impossible for a functional engine. An LSA of aligns the intake and exhaust lobes diametrically opposed to each other with no valve overlap at all. We need overlap so obviously neither situation will work for our needs.

While lift, duration and overlap are very important in the calculation of which cam profile to use in a particular situation, they do not go far enough for the thinking mechanic. Therefore, the manner in which the intake and exhaust lobes time to each other will give more needed information.

- In real life, Lobe Separation Angles usually range between and camshaft rotation. Again, one complete rotation is These are the most practical LSA's for most cam applications.

- Make no mistake; even though there is only a spread of about twelve degrees, relatively minor changes in LSA by even a few degrees will affect engine performance other factors being equal. A wide range of twelve degrees will have dramatic effect.

Let me stage a premise that will serve our learning purposes and compare a narrow LSA of and a wide LSA of to comprehend the effects of either.

Generalized Lobe Separation Angle Consequences

We can see from this extreme chart within the bounds of popular Lobe Separation Angles the dramatic effect narrowing

or widening a separation angle can have over another. The chart favors a narrower LSA choice. However, this is not always the case. Camshaft profiles must tailor to specific engines. A couple of LSA synonyms in mechanical jargon are Lobe Displacement Angle or the Lobe Spread. LSA measures in degrees of camshaft rotation and not degrees crankshaft rotation The camshaft rotates at half time of the crank rotation. The experienced mechanic will know this and can listen to the cadence of running engine components and easier isolate symptomatic noise. A narrow LSA favors off-the-line torque and power. A wider Lobe Separation Angle favors high speed cruising.

Camshaft Lobe Separation Angle Matters

From the above chart, it is easy to see that LSA is very important.

As important as lift and duration in cam choice is, we can take two cams with identical lift/duration profiles and yet one will perform better than the other will because of choosing a different LSA.

## Choosing a Cam

One should consider many factors before choosing a cam. You do not have to be a rocket scientist. You also do not have to understand fully all that we have gone through. However, you will have the knowledge to decide on a knowledgeable adviser.

Separate the marketing trivia from fact. Apply the available factual-information to your bike, total weight and riding style.

Cams for a Dyna will be different from those suitable for a Bagger.

One problem most riders encounter is that a radical cam costs the same as a mild one. Many will feel a need to purchase a more radical cam because it sounds like you are getting more lift, duration and overlap for the same amount of money. The truth is that you may be purchasing less for the same money because that wilder cam will bog down your ride.

## The Rule

The overriding rule is not to purchase a cam too radical for your bike or your needs. This is the number one mistake enthusiast and mechanics alike make. The bigger is always better syndrome does not apply to performance mechanics.

This condition seems to be endemic to the Harley rider. Everyone wants to get better bang for the buck. This is another factor in the go-big-or-go-home decision-making process that many times leads to less power because too radical a cam becomes the incorrect choice for a milder application. Why spend good money on a performance part that will not increase power and may even decrease what is already there?

Bigger is Better with big cubic inch engines that require increased duration and lift because of demands for more air, much more air.

Cam Duration, Compression and Fuel Modifiers

Modern stock EPA dominated engines need an increase in length of time the valves are open with a longer duration cam

for better breathing capabilities. With a longer duration non-EPA cam we can also increase the compression ratio because pressure begins to build later as the piston ascends the cylinder on the compression stroke because the intake valve is open longer.

Increased bore and stroke engines can absorb even longer duration cams. I only make this point for informational reasons since I will always increase compression whether the engine absorbs longer duration or not.

Increasing compression without equalizing duration will cause more heat generation, invariably leading to detonation. Therefore, an input fuel modifier is necessary to bring the air fuel ratio down to manageable levels. The ECM fuel tables link to the ignition tables so both modify with an input system like Terminal Velocity.

Power Commander is an output modifier that adjusts ECM signals after they leave the computer. However, the Power Commander has an ignition component that other output modifiers do not.

Larger cubic inches will also lower the power band created by the longer duration cam whereas the same cam in a lesser size engine will raise the power band.

Power-to-Weight Ratio

This subject is a biggie.

All things being equal, the lighter rider on a lighter bike with the same torque and horsepower will lose the heavier rider on the heavier bike.

Most people refer to power-to-weight as the relationship of engine power to the total weight of bike and rider. Power-to-weight really compares the power of the engine to the weight of the engine. Weight-to-power ratio, more correctly describes the relationship of engine power to the total weight of the bike and rider. As obvious as this may be, the reason is that the lighter weight has a higher power to weight ratio.

Is it time for a diet, anyone?

Power-to-weight and weight-to-power ratio also has a big effect on cam choices. The heavier ride will need a higher-torque, higher lift cam with shorter duration to move the power band lower in order the get things moving easier. Valve overlap, lobe centerline angles and lobe separation angles will also need consideration. If the heavier rider and bike does not care about low-end power because of riding at highway

speeds most of the time then longer cam duration will assist at these constant speeds and for passing at speed.

Lighter riders and bikes will move off the line easier so their cam of choice might move the power band into midrange or a higher range depending on wants and needs.

Compression Choice

The greater the compression an engine has just this side of detonating fuel mixture, the greater the power will be.

Detonation is the great engine destroyer. Many refer to detonation by other names like spark knock, knocking and pinging. Detonation evidences by sounds like ball bearings bouncing around in the cylinders. The cause is a spontaneous ignition occurring elsewhere in the combustion chamber

independent of the correctly timed sparked-ignition.

Each ignition event sparks the fuel and forms a flame front generating a pressure wave travelling across the combustion chamber. If pressure and temperature of the yet unburned mixture exceeds its ability to resist combustion the balance all goes off at once...detonates.

There are many causes for detonation. Some obvious ones are poor quality fuel, over-advanced ignition timing, overly high compression ratio, valve timing, rod-to-stroke ratio creating much higher piston speeds and the current lugging problem for some riders with 2006 Dyna and TC96 supercharged gear ratios.

Two much heat causes the pinging. Heat and pressure are synonymous terms when it describes burning fuel. The burning heat expands thus becoming pressure. Increased

pressures can spontaneously combust the fuel.

The Twin Cam generally makes 35-psi oil pressure at F at 2000rpm. As Wood says, “it is necessary to keep cylinder head temperatures under to to combat detonation.” 4

He recommends running the best oil cooler in both our opinions called a Jagg. In fact, Bob says to run two 10-core Jagg oil coolers in tandem to control cylinder head temperatures on a performanced Twin Cam. 5

I have always been a proponent of the use of oil coolers on Harley-Davidson’s since my early Shovelhead days. To me, the logic of using an oil cooler on an air-cooled engine is inescapable. I tried these dual 10-core oil coolers on my 10.5:1 compression Streetglide and as usual, Bob wood’s advice was excellent.

The major factor in raising cylinder head temperatures aside from out-of-whack timing is increased compression. However, we need more compression for power. We must compromise. Cylinder pressure increases with either an increase in compression ratio, which we can achieve in a variety of ways such as planning the cylinder heads, using higher dome pistons, thinner head gaskets, descending in altitude or a decrease in cam duration.

Cylinder pressure decreases with a decrease in compression ratio via thicker gaskets and/or a lower dome piston, increasing cam duration or gaining in altitude. Long duration cams need a compression ratio increase to maintain sufficient cylinder pressure to maintain or increase power. Gasoline quality and octane ratings have a huge limiting effect on maximum streetable compression ratios.

However, cylinder head design coupled with sophisticated engine management systems are allowing for higher compressions over time.

Hemi-head combustion chamber Panheads and Shovels used compression ratios in the neighbourhood of 7.0:1 to 7.5:1. These ratios were necessary even though utilizing lead as the excellent gas octane. However, the ignition was a relatively inefficient mechanical one using points and condenser.

Later Shovels from circa 1978 on used an electronic ignition that didn't get very good until the advent of the V-Fire III in 1983

Evolutions with flattop piston design and a D-shaped combustion chamber safely allowed a stock 8.5:1 with the V-Fire III, the beginning of an ever-increasing engine

management system.

Twin Cams began at a 9.0:1 compression ratio in 1999 using a flattop piston with a squish band extending right around the cylinder bore and an evolving sophisticated management system.

Performanced Twin Cams using refined electronic ignitions not hamstrung by emissions controls like the download-map Power Commander (www.powercommander.com), the fed, self-tuning Terminal Velocity (www.terrycomponents.com), and the IST (intelligent spark technology), S&S Cycle, Inc. (www.sscycle.com), proprietary engine management systems allow street compression ratios of up to 10.5:1 and sometimes beyond.

Other Cam Choice Factors

Other considerations to evaluate in cam choice are riding style such as touring or hotrodding to use two more extreme examples; bike weight also taking into account trailer towing or side hacks (sidecars); operational rpm range relative to gear ratios and the number of transmission gears, 3, 4, 5 or 6-speed. Yes, early Harley's sometimes only had three forward gears.

Engine displacement and compression ratios have a huge impact on cam choice. In fact, any modifications like high flowing air cleaners, air filters, carburetor size and jetting, EFI year as well as throttle body size, ignition or engine management system, compression, cylinder head modifications along with valve sizing, performance valve springs, and exhaust options all have small to great effect on cam choice.

When entering the world of super-aspirated NO2 (nitrous

oxide), and forced induction supercharging and turbocharging, well, cam profile design makes some diametrically opposed changes to normally aspirated (natural breathing), motors. I will delve into these last subjects in Chapter XIII: Forced Induction.

Rule II: Co-ordinating Performance Components

All the above factors hopefully will give some rather than take away from the cam profile design.

Coordinated and complementary parts choice is the goal. There is no place in this equation for superstars as performance can severely compromise without compatible mating components. Parts choice must be coordinated so each component complements the other, as a team that will perform better than the biggest superstar will.

Cam choice will become logical and harmonizing in this scenario rather than attempting to bring radically opposing components into a difficult contrarian fold.

## Valve Springs

### Twin Cam Gear Drive Development

In 1999, when installing hi-performance cams and mating heavier seat-pressure valve springs in the Twin Cam, leading performance mechanics simultaneously discovered that all the expected performance was not forthcoming.

The reason for installation of heavier seat-pressure-valve-springs in an engine is to ensure that the valve train precisely follows the dictates of the cam lobe profile. In other words, the valves open and close when instructed by the turning

elliptical cam lobe supported by stronger valve springs.

Performance springs mandate is to prevent valve float and valve bounce that robs horsepower and can demo an engine.

Valve float occurs when an errant valve hangs around in the combustion chamber after the cam lobe rotates to lower lift allowing the valve springs to begin closing the valve back onto its mating seat. High rpm or a wild cam profile with all lift and no duration can easily confuse the valve as the properties of momentum and inertia come into play. At the peak of the cam lobe, the valve will want to keep going, (momentum), to open further under these conditions. It may open further or oscillate (inertia), in no man's land until the valve springs take charge and force the errant valve back to its valve seat.

The cause of valve “bounce” the lesser-known cousin of valve float occurs when the valve hits it’s mating seat at high rpm. The valve tends to bounce off the seat causing much the same problems as float. Heavier seat valve springs will also correct this condition.

Valve spring pressure should closely match to an engine’s potential rpm range and the profile of the cam. It is a good idea to use the stock valve springs with stock or mild lift performance cams.

However, the stock cam timing chains and adjusters, particularly those used in 1999-2006, should replace with a pure gear drive produced by the likes of Andrews (www.andrewsproducts.com), Crane (www.cranecams.com), S&S or Wood Performance (www.woodcarbs.com).

Bobby Wood introduces the exotic leading edge

combination of an outboard carbon fiber belt combining with either spur or helical gear inboard gear drive's of the Knight Prowler Silent Belt Drive System when using higher lift cams requiring heavier seat-pressure valve springs.

According to S&S Cycle, Inc. and I agree is that the 1999 to 2006 stock timing chains and their mating chain tensioners are not strong enough to counter the extra seat pressure of performance valve springs.

Performance Beehive Valve Springs

In 1999, the Twin Cam continued to use the Evolution valve springs instituted in 1984. These dual cylindrical shaped valve springs (18201-83 outer and 18202-83 inner), one inside the other hold each valve closed on early Twin Cams up until 2003 and early 2004.

The performance version of these valve springs has been around since late 1984.

They supply in a variety of functional steel to exotic lighter and stronger material compositions like titanium and nitriding processes.

There is a wide choice of seat pressures providing matches to all cam profiles.

There is just as wide a choice with which manufacturers to choose.

The inner spring normally counter winds to the outer to combat fibrillation. The different sizes of inner and outer also combat this untoward spring movement away from designed compression and extension under duress of performance cam

design. The springs may also progressively wind, whereby coils are closer together at one end than on the other spring to prevent harmonics from impeding intended travel and function.

A single beehive shaped valve spring (18245-02), operates the closing of each valve on later Twin Cams from circa 2004 on. The valves also have narrowed diameter valve stems reducing valve and thus valve train weight.

Anytime valve train weight reduces, two beneficial results emerge; the engine can rev higher since there is less mass to move up and down; secondly a lower seat pressure will perform intended functions since there is less weight once again, to impede the valve's return to its seat. Less weight combining with an effective less seat-pressure-spring reduces valve seat slam from the returning valve.

The stock beehive spring-seat-pressure is only 130-lbs with the valve on its mating valve seat.

Compression steadily increases seat pressure.

Common dual seat pressures for Crane springs, as an example is 155-lbs seat pressure for milder performance cams and 175-lbs pressure for more demanding cam profiles.

Beehive springs, progressively change shape from narrow on the top valve collar to wide on the bottom lower collar to combat untoward harmonics. The changing diameters of the coils have progressively changing physical properties that battle fibrillation from gaining a foothold and expanding their counter-surging characteristics.

Interestingly, Screamin' Eagle (www.harley-

davidson.com), heads for circa 2005 and later applications that normally use the single beehive-spring substitute usage with dual cylindrical springs with seat pressures and clearancing to accommodate up to .650” lift cams, (stock lift is around .473”). It looks to me, like S.E. may not like its stock counterpart spring design.

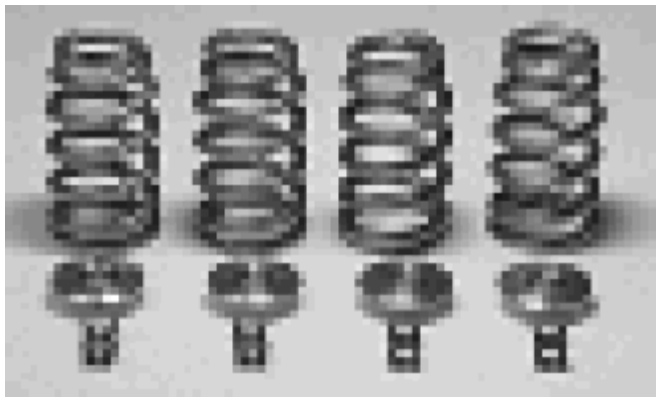
Bob Wood’s website, (www.woodcarbs.com), proudly proclaims on the subject of 2005 to present beehive valve springs, “Wood Performance has stepped up to the plate to give our customers the absolute latest and best kept secrets in NASCAR valve train technology, and incorporated it into our latest W160-PSW-7L KNIGHT PROWLER valve springs.” 6

The valves using stock beehive springs rotate inside their valve key locks. Valves have indents machined into them just below the top of the valve stem. The tapered valve locks, two

per valve insert into the mating valve stem indent with the valve spring compressed. Slowly release the top valve collar and spring until the mating taper in the top valve collar locks onto the keys, which are rounded. Each circumnavigates almost 180-degrees around the stem to complete the circle.

The Knight Prowler springs feature titanium seven-degree, 7mm posi-lock valve keepers and retainers. The Wood Knight Prowler springs have 160-lbs seat pressure at 2.000-inches compression; 360-lbs at 1.400” compression, 390-lbs at 1.300” compression and coil bind at 1.230” compression. Bob’s valve springs accommodate up to .600” lift and still maintain a very safe interference clearance of .060” without shimming or cutting of valve guides. They also feature an advanced 1200 series heat-treating process with an additional 1500 series nitriding.

Wood Performance Beehive Valve Springs



The Wood Performance Knight Prowler beehive W160-PSW-7L valve springs for all circa 2005 to present Twin Cams. The single valve spring utilized by the Factory after 2004 instead of previous dual cylindrical-shaped springs used before that actuates the closing of a thinner 7mm diameter valve stem lowering valve train weight allowing for slightly higher rpm and lower seat valve pressures. Stock seat pressure is 130 lbs while the Knight Prowler springs have an initial pressure of 170-lbs at 1.900 inches; 178-lbs at 1.875”

compression; 187-lbs at 1.850” compression; 368-lbs at 1.300” compression and they coil bind at 1.230” and accommodate .600” lift without shimming or cutting of the valve guides. Metallurgy is superior with advanced 1200 series heat-treating, titanium posi-lock valve retainers and 1500 series nitriding process.

Who Makes What

Bob Wood is very forthright about what he does, which is the design of both trick looking performance parts and kits but also of products that give loads of power and torque.

Bob’s Wood Performance exact proprietary specifications to leading industry companies result in stellar performance parts.

His cams are different in that they feature short duration, high lift specifications that have become a Bob Wood trademark; “They yield Very High torque and horsepower numbers, not matched by any other cams at this time.” 7

For example, Andrews Products makes Bobby’s Knight Prowler cams for him while K&N Engineering (www.knfilters.com), makes his high flow air filters. Actually, Andrews Products make all of the cams for different drive configurations that Bob designs from both early and late chain drive cams, standard gear-drive cam setups and of course, the grinds for the Advanced Belt Drive System with either spur gear or helical cam gears.

S&S Cycle, Inc provides their proprietary inboard cam gears and outboard drive spur gear-drive gears to power Bob Wood’s standard gear-drive cam kits.

Goodyear (www.goodyear.com), makes the hi-torque 5 mm carbon fiber belt to exacting specifications.

We have some pretty impressive and venerable American companies here that are industry leaders with their particular expertise. Bob is an all-American patriot who passionately believes in and uses American made products.

## Radical Cam Profiles and Engine Longevity

Over the years, I have been critical of radical cam grind profiles as being unreliable for engine longevity as they wear other valve train parts rapidly. I think it time to qualify my longstanding belief, which quite-frankly evolves from my Shovelhead and Panhead years. The other principle I should qualify is that I am a proponent of reliable horsepower and torque gains believing that engine efficiency and performance

many times marry.

I have watched a multitude of hemi-head; (hemispherical combustion chamber Knucks, Pans, Shovels and Ironheads) engines demo or wear out right before my eyes. The last statement is dramatic and of course, there is a sliding scale to its veracity directly proportional to the conservativeness or radical extremity of the performance options and parts used with these particular engines.

Premature wear and/or parts/engine destruction made life difficult for me in the early years. I would say; “If we do that, your engine longevity will suffer although the bike will be very much faster.” They would say; “No problem Don, make her real fast!” Foolishly, for me I would follow their instructions. Hey! They would loudly complain; rebuild my engine for free; it wore out too fast.”

Then the bad mouthing about my mechanical abilities would follow. I like to go into work every day, as it is my hobby as much as my vocation. I began to shy away from anything that caused me grief.

So, why did premature wear happen? Well, a few reasons; too high compression in the hemi heads, low by today's standards, resulted in detonation, the great engine destroyer, that the parts quality and technology of the day could not handle.

Extreme valve angles and pushrod angularity is another design feature of these yesteryear engines that were intolerant of low duration, high lift cam profiles. These engines needed longer duration cam lobes to ensure the lifters could follow them without flying off in a momentum, inertial driven frenzy promoting valve float, another great engine destroyer.

Then along came the Evolution engine, which is very much faster and reliable. Why is this? The head design could tolerate higher compression ratios while controlling spark knock relative to what went before.

Electronic ignitions fed by critical multi-sensor information began to modify ignition timing in a more exact and sympathetic manner than weak-spring loaded points mechanical ignition systems also allowing more but not enough control over detonation.

The biggie with the Evo was the reduction in valve and valve guide angles reducing side loading dramatically. Side load reduction, in itself, allows for cam higher lifts. It reduces valve guide wear and frictional loss of horsepower.

Moreover, rocker arm ratios increase to 1.625 in Evo and

Twin Cam engines from the Shovelhead 1.420 ratio allowing much higher valve lifts using the same cam lobe lift. Suppose we have a .290” cam lobe lift cam in both a Shovel and a Twin Cam; the valve lift in the Shovel will be 1.420 x .290” = .411” valve lift while the valve lift in the Twin Cam with the same 290” cam lobe lift will be 1.625 x .290” = .471” valve lift. The rocker arm ratio increases the valve lift by a full .060”.

Much-improved manufacturing techniques, newer technology and better quality materials further contribute favourably to an increase in compression ratios and higher lift cams.

Stock Panhead lift at .390” evolved into a .412” lift for the Shovelhead via an increased rocker arm ration suddenly skyrocketed to an amazing, for the day, circa .490 lifts in the Evo engine. A .490” lift cam in a Shovel was hairy but very normal in the Evo and Twin Cam.

Now we are getting somewhere.

However, the aggravating Evo and predecessor pushrod angles relative to the lifters remained a problem particularly with the front exhaust and secondly the rear exhaust. The center intake pushrods are much straighter. Even, the super-improved automotive style, strong Evo hydraulic lifters continue to fall victim to bleed-down because of the pushrod angularity.

With the advent of the Twin Cam, the valve angles and rocker arm ratios remain consistent with the Evolution engines. However, the location of the two cams allow almost straight up and down pushrods. Gone are the bleed down, resultant lifter ticking, and the inherent inefficiencies.

Certainly, radical cam lobe lift and duration figures in a Shovelhead become very normal and safe in the Twin Cam.

The Twin Cam engine with the improved descriptively named bathtub-squish-band-configuration head design again tolerates higher compression ratios over the single sided D-shaped squish band in the Evo. The Twin Cam is an efficient pushrod-operated engine given that it still has the restrictive inadequacies of a valve train. The better technology is overhead valve camshafts that provide direct and immediate actuating.

Metallurgy, robotic precise manufacturing and superior quality control vastly improve over the Evolution and are light years ahead of the Shovels and similar type engines.

In short, for an antiquated but beloved engine, we have huge changes and improvements that allow for radicalization

of previous standards as they now become normal.

The current sophisticated stock Twin Cam engine management system attempts better than anything before to protect against detonation. In fact, the ever-improving Delphi ESPFI (Electronic Sequential Fuel Port Injection) instructed by the sensor-serviced ECU (Electronic Control Unit) has a newer feature, the idle-temperature-management-system. In simple terms, it offers a cooling feature for hot idle conditions thus lowering cylinder head temperatures and the prospects of deadly engine destroying detonation.

Sophistication is beginning its reign.

The aftermarket has been hot on the trail with automotive crossover technology and new innovative designs. Exhaust system (oxygen), sensors feed information to the non-

emissions control engine-management-computers that interrupt and change input information to the ECM thereby changing the fuel-correlated ignition tables or maps. Pulse-width, which is the squirt time instructions to the fuel injectors, increases accordingly as do the stock piggybacking ignition-timing tables.

I digress as usual, but an input fuel-ignition aftermarket module like Terminal Velocity is superior to an output module that only modifies fuel delivery such as TFI (www.techlusion.com). However, the TFI does a good job modifying fuel delivery.

The output fuel modifier, Power Commander also has an ignition component but any ignition changes require a sophisticated technician. There are less competent people than one might think.

In any case, improving technology, metallurgy and computer sophistication is allowing, in my mind at least, the viewing of former radical cam design as an acceptable, safe and necessary component of current go-fast modifications. I would never have dreamed of going above an Andrews B-grind .485 in my 1968 Shovel ex-cop bike back in the day. I did try an Andrews C-grind, which has a .525" lift but I did not get any extra power because I failed to match other parts correctly, as most back then did.

We simply had not developed our knowledge levels sufficiently.

The problem is that a wilder cam cost the same as the appropriate cam for whatever performance configuration we were after. We were still learning the principle that there are no superstar parts but a coordinated complementary configuration of components, and furthermore, bigger is not

necessarily better.

I relented a little in my many Evos, rarely going above a .525 lift and occasionally try a daring .550 lift. However, I did not like the results provided by the higher lift cams better suited to an overhead cam profile because anticipated horsepower and particularly torque was not forthcoming because the power band was too high and out of range for normal street hotrodding. The power band was too high because the cam duration relative to lift was just excessively long!

I have changed my mind.

I tried the heretical to some .650 lift lower duration Knight Prowler cams because I talked to Bob Wood and trusted his judgement. I am having so much fun that I do not care if valve train engine longevity of a new Twin Cam is reduced

from say five or six years by a year or two.

However, I still would not go to these lifts in earlier engines like those that do. My pockets are not that deep. Of course, for those with money, it all reduces to more fun. For the knowledgeable, these are the best of all times, for the performance oriented H-D mechanic.

Summary Conclusion

Before I digress further, I bring to a close with what the Wood Performance website says better than I can, “IT’S NOT HOW FAST YOU GO… BUT HOW QUICK YOU GET THERE! TORQUE, TORQUE and MORE TORQUE is the ONLY way to put you there quicker and to “Smoke the Competition!...(This design and manufacturing process has) produced The Most Advanced, Aggressive camshaft profiles for H.D.…(Twin Cam)

Engines available anywhere!!...They yield Very High torque and horsepower numbers, not matched by any other cams at this time." 8

1 Bispham, Andrew. Email August 24, 2007 4pm.

2 Wood, Bob. Wood Performance Carburetors. Email, December 23, 2007 at 11.13 am.

3 Ibid.

4 Wood, Bob. Wood Performance Carburetors. Part of a telephone conversation early 2007.

5 Ibid.

6 Wood Performance Carburetors. Website, http://

(www.woodcarbs.com)/valvesprings.htm Knight Prowler Valve Springs.

7 Wood Performance Carburetors. Website, http:// (www.woodcarbs.com)/cams.htm Knight Prowler Cams.

8 Ibid.

# Chapter V

## Twin Cam Vibration

Harley-Davidson prohibits Twin Cam flywheel assembly rebuilds, re-truing or balancing and does not sell repair parts for them. Out-of-true flywheels now simply replace with a new set built from the Factory. Most H-D technicians today have never seen a set of Twin Cam flywheels apart. Their training does not include the science of making every rotating part of a crank true with each other.

Allowable flywheel truing tolerances today are sadly much more liberal than yesteryear.

I expound relentlessly regarding the necessary trueness of the engine's crank (flywheel assembly), as the mainstay of combating vibration caused at the very core of a Harley-Davidson.

Please forgive me as I continue to do so because I feel it so important.

Balancing the cranks' rotating and up-and-down reciprocating parts is a further refinement of truing the component parts together and in relation to each other that will help eliminate inherent vibration.

There are two main types of balancing called static and dynamic. Most mechanics favor dynamic balancing because

factors determine by spinning the flywheels more approximating actual engine running conditions. Only the venerable S&S Cycles, Inc. advocate static balancing, at least with their product…that is until recently reversing a decade's old policy. In my opinion static is good but dynamic is better.

I will go into much detail about balancing Harley-Davidson flywheels further in this chapter in subsection Engine Balancing.

I will do the same for truing Twin Cam flywheels also in the subsection Truing Flywheels.

## The Primary Cause of Internal Vibration

The primary internal cause of V-Twin vibration is the uneven spark-igniting firing positions of the angled cylinders. A four-stroke engine like Harley-Davidson requires of

crankshaft rotation to complete all four cycles or strokes. They are intake, compression, power and exhaust, each lasting crank rotation. One might think that each cylinder operates apart from the other. If the cylinders were this would be true.

However, we have cylinders. Therefore, the two cylinders fire awkwardly, but that is life in the wacky world of Harley, where original engineers felt a engine would not even run. One-cylinder fires at while the other fires later.

The Motor Company goes against the standard automotive industry`s contention that the rear cylinder is the master cylinder and therefore the number one cylinder. The front is number two slave cylinder. This is the logical way to look at this because the rest of the engine times off the rear cam lobes and the rear cylinder. However, the Factory's contrarian and illogical view is the opposite with the front cylinder being

number one with the rear following at number two.

At face value, this will seem logical and correct to the Harley rider as it initially did with me. I point out that the rotating pinion shaft's valve train drive connects via the primary timing chain to the rear cylinder camshaft-drive-gear. The rear cam connects and therefore, drives the front cylinder camshaft via the secondary timing chain.

With this information, what now could be clearer? The rear cylinder is indeed the number one cylinder with the front number two cylinder following its dictates. Again, Harley-Davidson uses the reverse. So, against the automotive industry`s inescapable logic, Harley-Davidson's front number one cylinder fires at and the rear number two cylinder fires later.

How can this uneven firing sequence do anything but

create unwanted vibration?

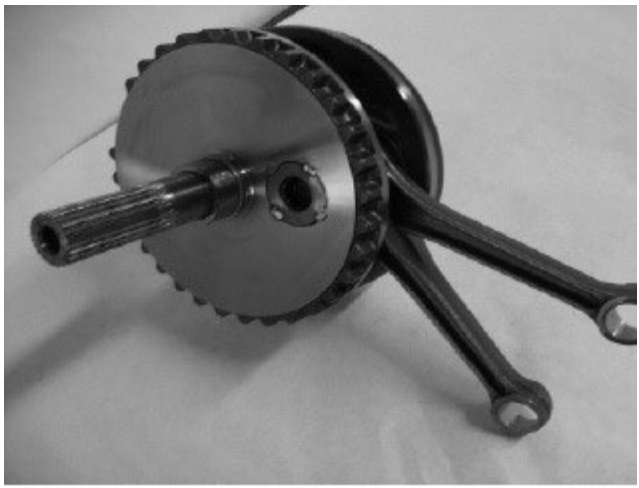
Uneven is the key word.

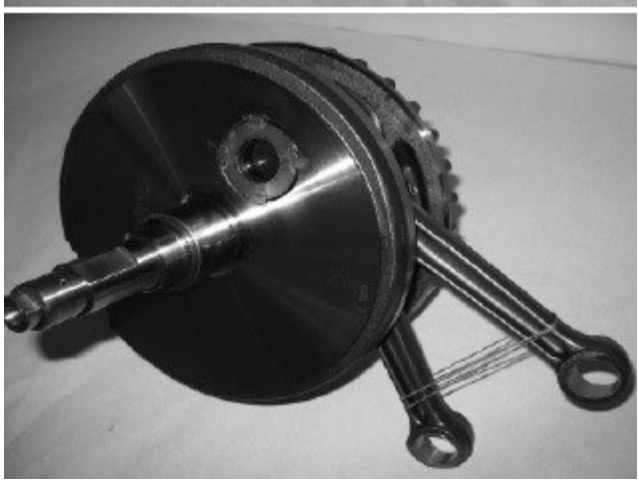
If the component parts of the crank are not true to each other exactly, this core vibration can exacerbate exponentially as in shake, rattle, and roll. I have stated, as have many others, my opinion repeatedly, that flywheel and shaft runout should not be more than one-thousandths of an inch (.001”). In actual practice, with modern flywheels, in many cases it is easy to get shafts less than .001”.

However, truing the flywheel rims to .001” is more difficult and may not always happen. I attempt to get the tolerance as low as possible but certainly never swillingly tomach more than .003”.

Furthermore, with precise, modern robotic machining in combination with an integral, perfect to flywheel component, it is not difficult for those proficient at the truing procedures to attain trueness in the ten-thousandths of an inch closer to zero (.0000”), than .001”!

Scissored Twin Cam Flywheels





Pictured is a set of 2007 to present Twin Cam flywheels. They are easily identifiable by the narrowing of the wristpin connecting rod small ends. TC96 and CVO TC110 con rod small ends taper from bottom to top so they appear pointed rather than squared like 1999-2006 con rod small ends. This reduces reciprocating weight of flywheel parts that go up-and-down as opposed to those that rotate like the flywheel halves. Reduction of reciprocating weight helps ease the effects of a ⅜" stroke increase from the TC96/CVO110 over the 4.000" stroke TC88 cranks. The pistons must go a further total distance of .375" travel in the cylinders but they must do it in the same time as the lesser 4.000" stroke flywheel pistons. This creates extra heat and duress on reciprocating parts thus the lessening of weight to offset these extrapolating effects. These flywheels are scissored. Scissoring occurs when one, usually the heavier sprocket-shaft-side power flywheel, or both flywheels rotate forward or back on their mating straight interference fit crankpin end. Thus, their alignment to each

other scissors away. This greatly affects flywheel trueness and shaft runout. I have seen a set where the pinion shaft was out-of-true by a full .025”. The scissoring in this case shattered the oil pump geroters and destroyed the cam support plate in and around the pinion shaft bottom-end oil-feed bushing. The threat of geroters shattering begins at .015” runout. The inside geroters sit on the rotating pinion shaft that drives them. The long flat on the round shaft near its middle is where the feed and return inner geroters mate with the shaft. If the reader looks at the smooth right-side pinion flywheel, towards the left side of the picture, it will be readily apparent that it is out of alignment with the indented rim of the sprocket shaft flywheel. Once oriented, the reader can follow the pinion flywheel forward around its circumference and further observe the scissoring in relation to the left side power flywheel. The sprocket side rim indents allow the magnetic crank sensor to relay positioning of the flywheels and thus every other moving part in the engine to the ECM computer. The ECM can then determine ignition firing of the spark plugs

at the correct spark timing for such factors as piston speed to ensure the timely and highest percentage of burn to drive the piston down on its power stroke. The cams and full valve train timing as well as piston positioning is dependent on flywheel rotational position. Flywheels may also be out of alignment in four other ways. The halves may cock sideways, parallel to each other directionally in tandem to the left or right; the flywheel halves may spread away from each other opposite the crank pin or they may pinch together opposite the crank pin. Scissoring puts the sprocket and pinion flywheel shafts out-of-line with each other from the crankpin. A measuring dial indicator will note high and low points on the shafts with scissoring. Shown in the next picture Sprocket Shaft Runout in the Engine in this chapter are both a dial indicator and an H-D manual indicator. In this case, the right side pinion shaft will move the indicator right or clockwise indicating the pinion shaft is higher. The sprocket shaft end is barely showing in this picture. A truing indicator on the left sprocket shaft would move to the left or counterclockwise indicating the

sprocket shaft is lower in this example. With cocked flywheels, the indicator will move right or clockwise showing that side-shaft tilted upwards depending on which way the flywheels cock. The other indicator will move to the left or counterclockwise showing that shaft tilting downwards. With spread flywheel halves, the indicator on each shaft will measure in unison with the other up or down recording high and low points in the same area of each respective shaft. Both truing indicators will move to the right or clockwise indicating upwards tilting of the respective shafts. With pinched flywheel halves, the indicator on each shaft will also measure in unison with the other up or down recording high and low points in the same area of each respective shaft. Both truing indicators will move to the left or counterclockwise indicating downwards tilting of the respective shafts. Truing flywheels is an art as much a science. It is very confusing at first. In fact, they confused me for years as I was learning. The concepts I describe here are applicable in 2008 just as they were a hundred and five years ago. It is also possible but rare

in my experience to have a bent shaft since they usually break, which is a separate and different situation. One shaft will be true while the other is out whack.

Flywheel Rim and Shaft Runout

One simple definition of runout is the variance of the circumference of a round surface relative to other mating round surfaces like opposing flywheel rims and shafts.

Flywheels

The flywheels press together ever since the advent of the Twin Cam in 1999 on the straight non-tapered crank pin. Previous Evos, Shovels, Pans, and Knucks use a tapered end crank pin that has threads on the ends. The threaded ends protrude through each flywheel face and the assembly holds

together via crankpin nuts. The sprocket-power-output-shaft casts together with the heavier left side flywheel as is the pinion shaft and the right side (pinion), flywheel.

All 2007 Big Twin sprocket shafts follows the 2006 Dyna design change in multi-splining the end where the compensator and alternator rotor package slides onto the sprocket shaft with their mating multi-splines. This replaces the fewer, ten-⅛”-wide splines in use on all Big Twin Harley-Davidson’s from circa 1955 to 2006. As it stands, this means that the 2006 Dyna and 2007 Big Twin 34-tooth compensator engine-sprocket components cannot retrofit to earlier Big Twins.

Nor can the earlier mechanism fit with the new multi-splined sprocket shaft.

Runout

Runout refers to flywheel truing specifications. I am currently as of this writing in March 2008, aware of runout on the pinion shaft of a full nine-thousandths of an inch (.009”), and know of the aforementioned CVO 110 with an unbelievable .025” runout from resultant flywheel scissoring.

At Heavy Duty Cycles (www.heavydutycycles.com), we find lots with .003” or .004” runout, which are more than one or two-thousandths of an inch (.001” to .002”), too much according to earlier 1999-2007 Twin Cam service manuals.

Incredibly, Harley-Davidson revises acceptable runout specifications dramatically upwards in the summer of 2007. In my opinion, these new runout specifications are at the point where parts wear really begins to affect the engine’s longevity. Of course, I prefer these measurements to be .001”

or less.

A specialized micrometer called a dial indicator measures accurate pinion shaft runout while in the engines cases. Removal of the timing cover and cam support plate is all that is necessary for this procedure. We perform this measurement every time while doing oil pump or cam work at Heavy Duty Cycles to ensure flywheel trueness. There is a great picture of this procedure in Volume I: The Twin Cam near the end of Chapter IV.

Importantly, no one has paid much attention to sprocket shaft runout with the flywheels installed in the engine cases. Moreover, why would anyone? The only part of the sprocket shaft that extends beyond the engine casing is splined whether it be the fewer and wider splines found on 1999-2006 (excepting the 2006 Dyna), Twin Cams or the more numerous

thinner splines found on the 2006 Dyna and all Twin Cams 2007 and later.

Well, guess what? In August 2007, the Factory revised acceptable runout of the sprocket shaft significantly upwards. They advise to change the sprocket-shaft-runout-specification in all shop workshop manuals from 1999 on to the more liberal ones.

In addition, H-D advises technicians on a procedure to measure sprocket shaft runout even with the interfering splines. Methinks that perhaps the sprocket shaft flywheel is sometimes coming out of true and scissoring relative to the pinion shaft flywheel. This spells for potentially disastrous consequences.

So, why is this happening?

Since the TC96 and CVO 110 engines are still relatively new, it may simply be a widespread bad production run. I am unaware of this being a problem with any TC88 flywheel assemblies. Perhaps, the Factory has reduced the press fit of the crankpin to both the left sprocket shaft flywheel and pinion right side flywheel downwards from the initial .007" interference fit. On the other hand, maybe, the increased torque generated by the TC96 over the TC88 is the culprit. It is probably a combination of both.

Why are the left flywheel affected and not the right? More correctly, either flywheel can scissor relative to the other but the potential is greater for the left-side flywheel. This question's answer is easy to understand. The much heavier left sprocket shaft flywheel is also known as the power flywheel because engine torque and consequent horsepower transfers out along the integral sprocket shaft to power the Drivetrain. The pinion side is lighter. It is along for the ride on light-duty

whose only responsibilities are powering the valve train and oil pump. Will sudden stopping or starting throw out the pinion side; sure, it's possible.

It bears repeating that each integral shaft is at a perfect 90-degree to its flywheel face through computer numerical control (CNC), machining. Truing one flywheel to the other follows after pressing the crank pin into each flywheel crank pin hole; anything more than a .003" runout of the shafts or flywheel rims will produce increasing excess engine vibration and premature wear of all affected parts especially the life-blood oil pump.

In my opinion, the pinion shaft bearing and the pinion bushing in the cam support plate will suffer along with negative effects on the all-important cam-support plate functions. The pinion bearing is huge and very resilient although longevity may compromise over time.

## Vulnerable Pinion Shaft Oil-feed Bushing

However, the pinion bushing in the cam support plate easily reams out larger if the much harder end of the pinion shaft is rotating out of true.

Why is this so important?

The oil feed to the all-important bottom end bearings, connecting rod big-end races and the crank pin channels through the pinion bushing into the pinion shaft for oil delivery. Pressure depends on a close tolerance of the pinion shaft where it passes through the cam-support-plate pinion bushing. A loose tolerance between the two reduces oil pressure to the bottom end components. Of course, there will also be readily observable oil pump inside wear, if there is a

problem with pinion runout. If disassembling an oil pump and wear is evident always, check pinion shaft runout first for the possible cause.

As I have said, a scissored flywheel will have more dramatic and immediate effect as it can actually break the pumping geroters themselves. The oil pump geroters are hardened steel while the oil pump body is a much softer aluminum. Broken geroters are an indication of severe trauma.

A scissoring of the right side flywheel by .015”, will likely initiate breakage of the geroters. Of course, the first indicator of this malady will be excessive vibration. The problem with noticing this symptom is the masking effects of rubber mounting or counterbalancing.

The second indicator will be zero oil pressure depending

on how far out-of-whack the flywheels become. The timing of each dramatic symptom may seem simultaneous as lack of oil pressure can follow vibration immediately.

Well, this subject lends itself to another Donny's Blasts from the Past.

Donny's Blasts from the How long will an H-D run without Oil?

Many years ago, I was biking to Canada's east coast with a riding partner appropriately nicknamed Harley from Toronto on our chopped Shovelheads, about a 1300-mile one-way trip.

As we rode on, two hundred miles into the trip, the return-oil-pump key sheared in my oil pump. I diagnosed this at 60 mph fairly-quickly because the engine was filling with too

much oil causing oil blowby past the rings into the combustion chamber. Resultant billowing blue smoke out my drag pipes and oil forcing out of the bottom engine casing vent splattering all over the rear of my chopper were the symptoms of too much oil in the bottom end of my engine. Sluggish performance is also a result of too much oil-drag on the churning flywheels.

As I gingerly slowed to a stop on my oil-soaked rear tire, the engine quit because of oil-fouled spark plugs. Harley and I pushed the Shovel into a deserted shed behind a gas station where we spent the next three days rebuilding the engine with tools we always carried and some Harley had to ride many miles to borrow. We did not really need to rebuild the engine but that was the mantra in those days.

Harley rode off with my heads 50 miles in one direction to recut the valve seats at some old automotive machine shop in some no name town. Then he rode 75 miles in another

direction to the closest H-D dealer to buy piston rings, gaskets and the all-important oil pump keyway.

On it went; Harley was getting in much riding in the blistering heat. In those days, Harley riders stopped every time if seeing another brother in the wind at the side of the road. Your riding partner hung tough and did what was necessary no matter the inconvenience.

There were neither cell phones nor the money to rent a crash truck or trailer. Bikers did not have credit cards because they had no credit.

At night, we would ride two-up to the nearest and only bar in Nowhere-Ville to shoot some pool and drink beer. We were dirty and skuzzy with greasy long hair and beards wearing the dirt and oil from rebuilding and sleeping on the asphalt.

There were no showers so there was no need for soap.

The locals did not like us.

As we left the bar, the second night, Harley's Shovel would not fire. Someone had stolen his spark plug wires. I knew what was about to happen. I had to back his play in the middle of nowhere, in the middle of the night without means of getting away. Oh, did I forget to mention the local-yokels outnumbered us big-time?

It was on; the ferocity of Harley's temper kept them at bay with me as his backup shadow as he crudely but effectively determined who had his plug wires. Plug wires in hand we made a hasty retreat before the crowd could regroup.

We blasted into the darkness, the straight pipes shattering

the nighttime air.

We slept on the asphalt another night until the hot sun woke us with our usual and quite normal hangovers.

I dumped a half-quart of oil into the bottom end over the bearings before installing the top-end. We greased or oiled all internal parts during assembly. Finally, my 66-Shovel was purring like a kitten. Unbeknownst to us, the oil-pump-feed key broke on initial startup.

I did not ride the bike hard because of the necessary break-in procedure.

So how far will a Harley engine run in very hot summer temperatures in this condition? At about 200-miles the bike began to feel sluggish. Heat was wafting off the engine…but

the day was hot. Around 225-miles, my bike began to labor. At about 240-miles, my bike began to slow as I gave her more gas. The poor old girl slowed to a seized stop.

Ahh, the life of a biker was not always easy. Looking for adventure and whatever came our way was not always fun.

Well, let us get back to the subject at hand.

Sampled Runout Specifications

In general, the newer the H-D engine, the more precisely made are the parts components of the flywheel assembly. Yesteryear production involved individual machinists on their antiquated lathes and milling machines making or modifying one component part after another. Worn tooling aggravated variances between machined parts.

I have seen Pan and Knuck flywheel sides vary by as much as an eighth-inch in diameter. Just try to true a mating set of different sized flywheels. Science is out the window and art takes over full time.

The Factory has been rapidly modernizing tooling, robotics, and quality control for a couple of decades now. It is now rare to find flywheel halves varying in thousandths-of-an-inch.

However, published specifications outlining H-D's runout limits are much more liberal. Rubber mounting Baggers and Dynas while counterbalancing the Softails shields the rider from vibration. I feel that H-D specifications are slacker now because of this vibration masking.

With rigid mounted, non-counterbalanced engines, the

Factory tried to keep specifications much tighter because vibration travelled right to the rider's hands, feet, and butt.

The 2007 Touring Manual instructs to replace the flywheel assembly if runout at the flywheel rim exceeds fifteen (.015") and/or runout at the flywheel pinion or sprocket shafts exceeds three-thousandths of an inch (.003"). 1 The 2002 Softail Manual teaches that the flywheel assembly runout at the flywheel rims should be between zero and ten - thousandths of an inch (.000"-.010"), and runout at the flywheel pinion or sprocket shafts should be between zero and two-thousandths of an inch (.000"-.002"). 2

A 1980 Shovelhead rubbermount FLT manual limits flywheel runout at the rim to no more than .006" and a maximum of .002" at the These 1980 specifications are shocking for me, as I never realized this liberalization of

specifications before. This is the first rubber mount Harley-Davidson, three decades ago. The purpose of rubber mounting is to dampen vibration. Yet, flywheel-truing tolerances relax to produce more vibration!

I can only think that specifications are loosening and therefore, sloppy because of masking rubber isolation. This enigma will be a continuing theme as I go on. This riddle reaches a crescendo in the summer of 2007.

But first, the Duo-Glide/early Electra-Glide manual for late rear-shock swingarm Panheads and early Shovelheads from the late 1950's and through the 1960's to 1969 has the same close specifications of .003 on the rims and .001" on the Let us go back even further to the rigid frame era for the Panhead Service Manual: 1948-1957 Rigid. These babies were the real boneshakers. Again, their specifications match those of the aforementioned Duo-Glides and

1999-2009 Engine Assembled: Maximum Allowed Flywheel and Shaft Runout

NA = not applicable

* Corrected H-D specifications for 1999 to 2007 Twin Cams as well as years 2008 and 2009 are runout of the flywheel sprocket shaft measured in the crankcase should be between .010” with replacement of the flywheel assembly to occur at .012”. Harley-Davidson feels that 1999-2007 pinion shaft runout maximum figures of .002” or .003” depending on year should increase to .010” and replace at .012”. These figures also apply to back to 1999 for the sprocket shaft runout in the

engine casing although there was no procedure for this before 2007. Sprocket shafts from 1999-2006 are more difficult to read because of fewer and thus wider sprocket shaft splines although a careful eye will eliminate the bumps to see approximate runout. I also re-true Twin Cam cranks rather than expensively replacing them. Tig spot-weld the crank pin ends to the flywheels in four equidistant spots with stainless steel rod. The Factory relays in its TT324 –August 22, 2007- Twin Cam Crankshaft Inspection & Run-Out Specifications circular that in general, riders will not feel crankshaft runout until .020" (twenty-thousandths of an inch), runout! 6

**These are my acceptable runout measurements whereby I follow the old specifications. What was good in yesteryear engines is certainly just as good today. In fact, these old tighter specifications should be much easier to attain today because of the integral shafts with their mating flywheels, CNC robotic machining, and superior metallurgy. Just because rubber mounting or counterbalancing an engine masks

vibration is no reason in my mind to loosen tolerances

because vibration reduces parts life, whether masked or not.

Engine Disassembled: Maximum Allowed Flywheel and Shaft Runout on Truing Stand

*Pinion shaft and sprocket shaft tolerances now measure more accurately at the accessible bearing races. Harley-Davidson feels that 1999-2007 pinion shaft runout maximum figures of .002” or .003” depending on year should increase to .000” to .004” and replace at .005”. These figures also apply to back to 1999 for the sprocket shaft runout. I also re-true Twin Cam cranks rather than expensively replacing them. Re-truing will probably disrupt the press fit of crank pin to flywheel halves. Do not trust re-trued Twin Cam flywheels unless restoring the rigidity integrity of the crank pin press fit. Do this by Tig spot welding the crank pin ends to the flywheels in four equidistant spots with stainless steel rod. Grind weld high spots without disrupting actual weld.

**The 1980 FLH specifications are tighter than the 1980 FLT ones. In my opinion, H-D cannot get away with looser

tolerances in the rigid mount FLH Shovelhead, as the vibration will increase dramatically transferring to rider hands, feet, and butt. FLT rubber mounting will mask the extra vibration from the increase in allowable specifications. The vibration will still be there but the rider will not feel it.

***These are my dream specifications that I always aim for no matter the year. It is not always possible to get rim runout down to .001” but for me, anything beyond .003” at the rims is intolerable. Decreasing vibration through closer tolerance specifications for runout is about more than rider comfort. It is also about parts longevity. Increasing vibration proportionately decreases engine life.

Sprocket Shaft Bearing(s) Endplay Specifications for Flywheel Lateral Movement

A is the acronym for the Alpha FLT Twin Cam Bagger and Dyna rubbermount engine while B is the acronym for the Beta counterbalanced Twin Cam Softail engine.

*Nineteen forty to 1955 pinion and sprocket shaft bearings comprise of loose rollers in take-apart cages. Different thickness thrust washers establish endplay. In these years, flywheels floated sideways in the engine casings. Endplay tolerance specifications prevented the floating flywheels and connecting rods from hitting stationary parts. The 1999 to 2002 Twin Cams use OEM 9028 tapered Timken bearings and races with an endplay spacer between the opposed bearings. This system, in use since 1956 and discontinued after 2002, is excellent in every way. A roller bearing substitutes for the Timken bearings beginning in 2003 to present day. Endplay

adjustment is precise in the earlier Timken bearings. The later roller bearing will fit at around .005” endplay when new in my experience. As the bearing wears from the stresses of sideways power pulses along the sprocket shaft, endplay increases. Counterbalanced Softail models have similar if not the same specifications except for flywheel endplay “float” inside the crankcases. There is less room because of the counterbalancers so the endplay tightens up considerably. Maximum endplay for rubbermount models is ten-thousandths of an inch (.010”) or .254 millimeters (mm), which in my opinion is too sloppy. I spoke too soon. H-D increases maximum endplay to .013” or .370mm in 2009. At Heavy Duty Cycles, we prefer to set endplay specifications on engines we rebuild to the lower limit of three-thousandths of an inch (.003”) or .080 mm. As an example the flywheel assembly endplay specifications for a 2002 Twin Cam Softail is a minimum of one-thousandths of an inch (.001”) or .030 MM to a maximum of five-thousandths of an inch (.005”) or .130 MM.

Flywheel Runout and Endplay Specifications in Millimeters (mm)

NA = not applicable. Harley-Davidson has stopped giving acceptable and replacement tolerance specifications in 2008. Personally, I find this intolerable. I also do not like what I feel are excessive flywheel rim and shaft tolerances now going back to 1999.

* These are my dream specifications that I always aim for no matter the year. It is not always possible to get rim runout down to .0254mm but for me, anything beyond .0762mm at the rims is intolerable. In the chart, I say sprocket shaft runout while the engine is assembled should not be more than .127mm but this figure is mostly meaningless for me because

measuring runout at this location is really all about diagnosing whether the flywheel halves have scissored, one in front of the other. Decreasing vibration through closer tolerance specifications for runout is about more than rider comfort. It is also about parts longevity. Increasing vibration proportionately decreases engine life.

## Fixing the Symptom Solves Nothing

From observing the above comparative charts, it seems that while materials and machining have dramatically improved over the years, the specifications for runout have slackened at the same time.

The fix-the-symptom not the cause approach with counterbalancers and rubber mounting allow this. What is the reason for this approach? Harley does not feel vibration is a perceptual problem to the rider until crankshaft runout

exceeds .020”. 7

Wow!

I am not disputing what the Factory says here but why are we even going there? When the Milwaukee Vibrators reigned like the rigid Panheads maximum runout could not exceed .003” at the rims and .001” at the shafts. In my opinion, internal vibration will become progressively excessive above .002” shaft runout. In addition, reduced engine longevity, specifically to the cam-support-plate-bushing and the heart of the engine, the oil pump will begin at .003”. Accelerated parts damage will certainly begin to occur at .005” shaft runout.

Now here is the kicker that started me on this article.

H-D Backdating Corrected Runout Tolerances

The Factory, in August of 2007, states that the runout specifications published in all Twin Cam manuals from 1999 to 2007 need correction! The new corrected specifications are runout of the flywheel sprocket shaft measured in the crankcase should be between .000” to .010” (zero to ten-thousandths of an inch) with replacement of the flywheel assembly to occur at .012” (twelve-thousandths of an inch). 8

Aftermarket flywheel assemblies from the likes of JIMS (www.jimsusa.com), and S&S Cycle, Inc. (www.sscycle.com), rarely if ever exceed .001” runout with many measuring in the tenths-of-a-thousandths of an inch. I have installed some with no apparent runout at all according to my Harley-Davidson truing stand using dial indicator micrometers.

Scissoring Flywheels

The Factory is now indeed encountering many problems beginning in 2007 and carrying over into 2008 with flywheel halves scissoring out of trueness alignment on the TC96 engines and the CVO 110 cubic inch mills.

This blows me away because I know how rigid and together TC88 cranks are. There are two possible scenarios here other than incorrect factory initial assembly already discussed. One, the increased torque of the TC96 and especially the CVO 110 over the TC88 is causing this malady. In fact, this is the case whether alone or in concert with another factor. I cannot help thinking that the Factory for whatever reason, maybe an easier and therefore more cost effective truing process, has reduced the .007" press interference fit to a lower value.

Harley-Davidson warns repeatedly that Twin Cam flywheel assemblies are not rebuildable. The Factory enforces this

dictate through warranty rules and not making available any rebuild component parts. There are those of us who like a challenge and love to rebuild the un-rebuildable.

I understand scissoring flywheels, because if the power side torque becomes too great and/or the straight-interference-fit crankpin compromises then one flywheel moving ahead of the other is a no brainer. Scissoring occurs when one flywheel, usually the heavier sprocket-shaft-side power flywheel, or both flywheels rotate forward or back on their mating straight interference fit crankpin end. Thus, their alignment to each other scissors away. This greatly affects flywheel trueness and shaft runout.

The aforementioned flywheels where the pinion shaft was scissored out-of-true by a full .025” shattered the oil pump geroters and destroyed the cam support plate in and around

the pinion shaft bottom-end oil-feed bushing. The inside geroters sit on rotating pinion shaft that drives them. The long flat on the round shaft near its middle is where the feed and return inner geroters mate with the shaft.

If the reader views the picture of the scissored flywheels, the smooth right-side pinion flywheel, towards the left side of the picture is out of alignment with the indented rim of the sprocket shaft flywheel. Once oriented, the reader can follow the pinion flywheel forward around its circumference and further observe the scissoring in relation to the left side power flywheel. Scissoring moves both, the flywheel shafts, sprocket, and pinion, out-of-line with each other from the crankpin.

A measuring dial indicator will note high and low points on the shafts with scissoring.

Shown in the next picture, Sprocket Shaft Runout in the

Engine in this chapter is both a dial indicator and an H-D manual indicator In this case, the right side pinion shaft will move the indicator right or clockwise indicating the pinion shaft is higher.

The sprocket shaft end is barely showing in this picture. A truing indicator on the left sprocket shaft would move to the left or counterclockwise indicating the sprocket shaft is lower in this example. The runout measuring procedure with the crank in the engine cases should be done on the finer sprocket shaft splines, in 2006 Dyna and 2007 and later Twin Cams, noting runout variances on the tops of the splines and ignoring those caused by the spline valleys.

The attaching end nut certainly will not be concentric and other areas where measurement is possible may or may not be accurate.

My position is that the sprocket shafts are concentric at the ends because it is part of the manufacturing machining process. My opinion wins thus far, as all I have tested prove to be concentric. However, my opinion should not prevent you or me from checking this each-and-every time.

The reason is that before the hardening process, the machining centers are indeed accurate. With hardening, it is possible that the centers will distort minutely.

We found runout on one set of flywheels at the end of the sprocket shaft, on the fine splines to be .001” using the dial indicator micrometer. I thought this might be inaccurate since Steve; my head wrench had .001” runout on the sprocket shaft bearing surface beside the flywheel half also. This should extrapolate into a higher figure the further out the shaft to its

extremity it measures. Steve removed one of the two Harley-Davidson truing stand indicator bars and jury-rigged another mount with vice grips. Sure enough, the dial gauge micrometer was accurate, confirming .001” runout at the end of the sprocket shaft as well as on the bearing surface.

The rule is always to verify the shafts for concentricity.

## Other Types of Twin Cam Flywheel Misalignment

The Factory in some service bulletins notably TT324, Twin Cam Crankshaft Inspection & Run-Out Specifications on July 26, 2007 and TT352, Twin Cam Crankshaft Assembly Run-Out Inspection on December 3, 2007 along with two short videos are teaching technicians about flywheel alignment boondoggles.

Some of these young people have never even trued a set of

flywheels before or taken apart a Twin Cam crank.

The service bulletins say that traditional tapered crank pin and shafts out-of-true conditions can exist with Twin Cam Therefore, flywheels may also be out of alignment in four other ways than scissoring.

I do not get this.

The halves may cock parallel to each other directionally in tandem to the left or right; the flywheel halves may spread away from each other opposite the crank pin or they may pinch together opposite the crank pin.

With cocked or pinched flywheels opposite the crankpin, the indicator will move right or clockwise showing that side-shaft tilted upwards depending on which way the flywheels

angle. The other indicator will move to the left or counterclockwise showing that shaft tilting downwards.

With spread flywheel halves opposite the crankpin, the indicator on each shaft will measure in unison with the other up or down recording high and low points in the same area of each respective shaft. Both truing indicators will move to the right or clockwise indicating upwards tilting of the respective shafts.

With pinched flywheel halves, the indicator on each shaft will also measure in unison with the other up or down recording high and low points in the same area of each respective shaft. Both truing indicators will move to the left or counterclockwise indicating downwards tilting of the respective shafts.

Truing flywheels is an art as much a science. It is very

confusing at first. In fact, they confused me for years as I was learning.

The concepts I describe here are applicable in 2008 just as they were a hundred and five years ago.

It is also possible but rare in my experience to have a bent shaft since they usually break, which is a separate and different situation. In this case, one shaft will be true while the other is out-of-whack.

So, the flywheel-halves can apparently cock inwards as well as spread outwards opposite the crank pin. Furthermore, Factory service bulletins indicate the flywheel halves can slant to the left or right in tandem on their mutual crankpin ends!

I personally did not believe this possible because of the

massive diameter sizing of the straight crankpin that is hardened and fitting into straight holes.

Lastly, the flywheels can scissor where one flywheel moves ahead of the other. Understanding scissoring is a no-brainer. One flywheel will move forward or backward on the crankpin.

Appreciating the first four inward, outward plus left and right slanting conditions are easy on predecessor flywheel assemblies where shafts and crank pin have male tapered ends that fit into female tapered holes. The flywheels can easily shift inwards, outwards or skew in tandem sideways on their tapers. The separate individual pinion and sprocket shafts of yesteryear may also skew because of dirt in the flywheel taper, burred locating Woodruff keys or mechanic inattention.

Nonetheless, how does a straight hardened Twin Cam crankpin allow inward and outward shifts or tandem skewing

when either male end interference fits into smaller straight holes in the flywheels? Even though Harley-Davidson instructs in every Twin Cam manual to replace the complete crank in the case of misaligned flywheel assemblies because it is not possible to re-true them, we have come up with a procedure at Heavy Duty Cycles to do exactly that.

I had a mental blockage. I felt it impossible that the Twin Cam flywheels could misalign except in a scissoring situation.

Well, I was

We started checking the trueness of every set of Twin Cam flywheels as we have done with every other Factory crank for the last 35-years during any repair work. If this does indeed occur, it seems to me that it could only happen in the initial assembly process at the factory.

Sure enough, we have found every condition including pinching, spreading, slanting and of course, scissoring.

How does Twin Cam Flywheels Misalign?

First, the flywheels and shafts are CNC machined. Crankpin holes are similarly bored. The manufacturer grinds crankpins round. In my opinion, any fault does not lie with mal-machined parts. H-D tooling as well as super precise light-measuring techniques at the Factory is superlative.

I think pressing the crank pin under great pressure off-kilter into its mating smaller hole would be necessary. Material would unavoidably shave off on side of the hole to allow a misaligned fitment. If this were the case, compromising the .007” press interference could easily allow scissoring of the flywheels to follow.

One would think that the straight fit of the crank pin into each flywheel half would not allow this.

Another possibility is that a metal shaving or other piece of foreign debris might lodge inside the flywheel crankpin hole and skew the flywheel when the pressing operation takes place. Furthermore, many times a surface will be perfectly flat such as a gear face or concentric as in the round and straight crankpin before the hardening process. During hardening, the gear face may develop irregularities or bumps. H-D uses a grinding technology to resurface the gear faces afterwards.

It is likely that irregularities may occur on one or both crank pin ends where they press into the flywheel halves. This will certainly skew the flywheels into an untrue positioning as they press together. The .007” interference press fit may be

less in some cases allowing scissoring and other misalignment occurrences. Lastly, maybe the sheer volume of production is leading to mistakes in the pressing and truing of the flywheels relative to the crankpin.

Alternatively, there may be a massive bad production run.

## Twin Cam Crank Rebuilding Problems

First, once the flywheels are disassembled and repressed together the interference fit will not be the prescribed .007 press fit. There will not be a tight enough fitment as some mating metal is scraped away both in the removal and the reassembly process.

Other than resorting to a completely new crank, there are two main ways of dealing with this lack of rebuild specified interference fit.

One is the usage of adhesives; space age adhesives. There are Loctite's and other chemical adhesives, when used will more than make up for a few thousandths of an inch. The concept is easier to grasp if visualizing the words chemical

Re-pressing the flywheels together on the crankpin ends needs the assist of heat expansion and cold contraction. Even a .005 press interference fit will peel metal away during the press process if mating component parts were at room temperature. The crank pin needs to contract in diameter sizing by cold; perhaps even nitrogen cold, while the flywheel crank-pin-holes are heat-expanded.

Another area needing addressing is how do we know the exact sizing of the disassembled Twin Cam flywheel's crank pin hole and if the hole is perfectly rounded? We measure it

of course. If out-of-round, we will have to re-bore the hole to a round fitment size with precise tooling available in any machine shop or sophisticated mechanical service department. We cannot go too big or we will have to replace the flywheel half. For the technically proficient, this procedure will be simple.

Heat and cold fitment is again necessary to prevent metal peeling during the pressing process.

Truing the flywheels is the next issue requiring fast thinking, fast-gloved hands, and lots of experience. The flywheel rims require runout of .000” to .003”, and hopefully a lot less than my reluctant maximum. Shafts must be .000” to .001”, hopefully in the tens-of-a-thousandths of an inch.

Flywheel rims may be closer or spread from each other on

one side than the other one. They may also be out of true and cocked forward and back relative to each other. Check runout both at flywheel rims and at or near the ends of the sprocket and pinion shafts.

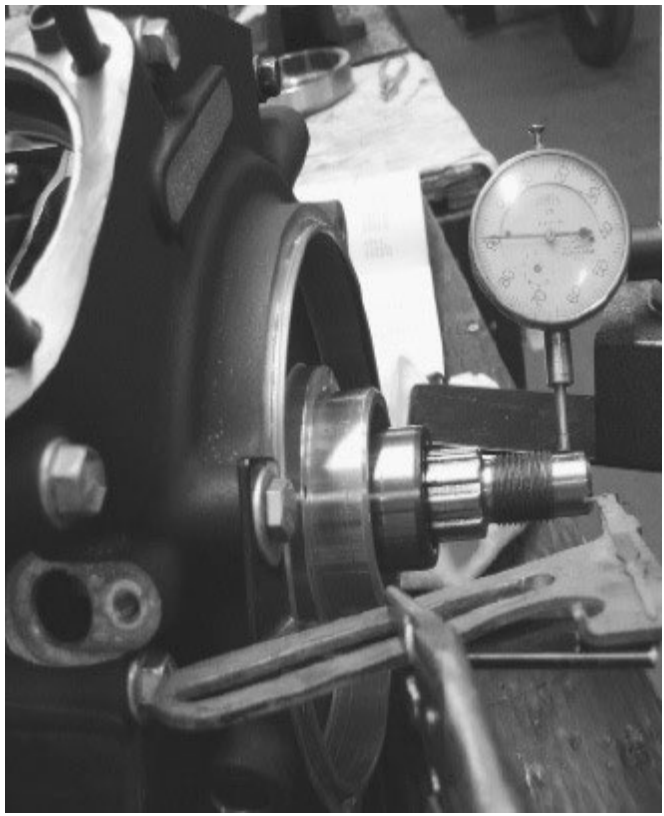
Vibration occurs more at the end of the shafts because measurements on the shaft bearing surfaces near the flywheel halves are less. Runout at the bearing surface will extrapolate to greater runout at the end of the shaft.

Modern technicians generally replace component parts assemblies after reading electrical gauges, consulting diagnostic charts, and obeying the diagnostics of the central computer in Milwaukee. Their training moves away from rebuild techniques.

In 2007, my curiosity and abhorrence of passing on faulty information got the better of me so I began disassembling

problematic Twin Cam flywheel sets to see what was going on. I also began to experiment with re-truing techniques. The techniques proved to be the same as in the past so re-truing the Twin Cam wheels requires the same skills as in yesteryear.

Twin Cam Sprocket Shaft Runout in the Engine



Pictured is an installed set of 2003 to 2006 Twin Cam flywheels in their engine casings with .0045” endplay. I mention this because the engine had .010” endplay before our repairs. The roller bearing sprocket shaft bearing varies between the H-D recommended .003” to .010” over the Twin Cam years. Maximum endplay for rubbermount models is ten-thousandths of an inch (.010”) or .254 millimeters (mm), which in my opinion is excessively sloppy. At Heavy Duty Cycles we prefer to set endplay specifications on engines we rebuild to the lower limit of three-thousandths of an inch (.003”) or .080 mm with the early 1999-2002 tapered Timken (OEM 9028), assemblies while the Factory allows up to .006”. We set roller bearing endplay from 2003 to present to a maximum .005” when using the 2003 to present series roller bearings (OEM 24604-00, 24604-00A, 24604-00B, 24604-00C, 24604-00D), although as I have said the Motor Company allows up to .010” or .254 mm on rubbermount models. In 2009, specifications relax further to .013” maximum endplay. Counterbalanced Softail models have

similar flywheel and shaft runout specifications as the rubbermounts but allow less flywheel endplay “float” inside the crankcases. There is less room because of the counterbalancers so the endplay tightens up considerably. As an example, the flywheel assembly endplay specifications for a 2002 Twin Cam Softail is a minimum of one-thousandths of an inch (.001”) or .030 mm to a maximum of five-thousandths of an inch (.005”) or .130 mm. I am going backwards here because we already have re-trued the Twin Cam flywheels to within acceptable limits for myself and many others in the industry, which is to .003” runout at the flywheel rims and .001” at both pinion and sprocket shafts. Once I set endplay for the roller bearing at .0045”, Steve, my head wrench for the last 28 years and I decided to double check runout at the end of the sprocket shaft. Now, H-D feels that this is not proper protocol because the sprocket shaft end may not be concentric, which will give false readings. My position is that the sprocket shafts are concentric because it is part of the manufacturing machining process. At the least, in this

particular case, my opinion wins. We found runout at this extremity to be .001” using the dial indicator micrometer. I thought this might be inaccurate since Steve had .001” runout on the sprocket shaft beside the flywheel half. This should extrapolate into a higher figure the further out the shaft to its extremity it is measured. Steve removed one of the two Harley-Davidson truing stand indicator bars and jury-rigged another mount with vice grips (partially shown). Sure enough, the dial gauge micrometer is accurate, confirming .001” runout at the end of the sprocket shaft. We trued these Twin Cam flywheels having found them to be scissored, one in front of the other, as well as the flywheel halves too close or sandwiched to each other opposite the crankpin location. The Factory says not to true the flywheels but to replace them if out of recommended specification tolerance. The problem is that a new set may also be out-of-whack relative to my personal the same-as-always maximum runout of .001”. Corrected H-D specifications for 1999 to 2007 Twin Cams as well as year 2008 are runout of the flywheel sprocket shaft,

measured in the crankcase, should be between .000” to .010” with replacement of the flywheel assembly to occur at .012”. 8 It would be wonderful to buy a set of H-D Twin Cam flywheels with a .000” runout at the sprocket shaft. First, this trueness is possible as I have seen this with a sprinkling of aftermarket flywheel sets but I have not yet seen a set of H-D wheels this true. I agree that if re-truing the straight-fit crankpin attached flywheel halves that the interference fit will be disturbed and the flywheels prone to going out of true even more once the torque of the churning flywheels enters into the running equation. That is why our machinist, Marvelous Tig weld tacks the crankpin pin to the flywheel in four equidistant spots with stainless steel rod.

## Survival of the Fittest: A Brief History

### The Japanese Threat

Harley-Davidson's survival-expansion into new untapped segments of the marketplace necessitated the development of a tamer product. This gentler realization for H-D began to percolate in the mid-seventies. Expansion by dramatically increasing unit sales would bring necessary modernization and economies of scale.

Although sad to say, I feel the Japanese had a technologically better product, was more reliable, cost less money, and most certainly vibrated less.

Eventually, Harley-Davidson could effectively compete with the invasion of the Metric V-Twins with resultant improved quality and decreased pricing. Look to the year 2008 and after to see the results of H-D's efforts. The Factory is meeting Japanese competition successfully on every level, particularly with aesthetics, quality and pricing. Did I leave

out comfort?

H-D Dominating the Aftermarket

The second prong of the Factory attack to ensure dominance of their marketplace involved attacking the aftermarket on three fronts: customization, performance, and aesthetics…with chrome etc.

When Heavy Duty Cycles began business in the early seventies chrome, paint, rebuilds and cam-work were the major parts of our business. Now, factory Harley's come chromed. For those extra pieces requiring chrome, well, they are now available at the dealer. H-D paint has improved so much that I rarely paint my touring bike anymore, although my city chopper still receives custom paint.

Harley-Davidson increased reliability, horsepower, and

torque also take a bite out of the aftermarket. Just look at the V-Rod series. The V-Rod models come in many exciting configurations now to serve many tastes. Prices have dropped dramatically.

The TC96 using a 4 ⅜" stroke and a 3 ¾" bore is now standard with an easy upgrade to 103 cubic inches by safely increasing cylinder bore to 3 ⅞".

To those of us that have been around for a few decades, we know that the S&S 96 cubic inch stroker with a stroke of 4 ⅝" and a bore of 3 ⅝" was God to the performance oriented of the day. The S&S 96-incher was pretty much the limit for a sort-of-safe big inch Harley. It was the safest big engine. Sure, lots of us ventured into 4 ¾" strokes, which took the Evo or Shovel out to a whopping 98-inches but reliability was not as good. The engine cases of the day were not well equipped to

handle much more than a 3 ⅝” bore.

At Heavy Duty Cycles, we routinely build S&S 116.2” (4 ⅝” stroke and a 4.000” bore), and 123.6” (4 ⅝” stroke and a 4 ⅛” bore), engines using stock Twin Cam cases. The 124” engines we build today are more reliable than the much smaller 96-inchers of yesteryear. In fact, they win hands-down.

As for customization dominance, check out the 2008 and later Softail Rocker, the Dyna Fatbob series and the new V-Rod’s over the last few years. Never mind that, let us look at the base model for the Dyna. The Factory has included a fatter rear wheel and tire, an aircraft gas tank cap and V-Rod fat fork tubes to trump the skinnier 41mm wide glide forks, customizers were using.

The price did not go up; it went down!

However, before any of this could happen, the beast needed some domestication. The old Milwaukee Vibrators were acceptable to the hardcore riders of old, the military and of course, the police departments. The hardcore were the only ones who embraced these wild steeds, as the riding police officer and the army recruit had no say in the decision-making.

However, only the hardcore, mostly the outlaw subculture, were unswerving in loyalty. Other previous steadfast markets like some police departments were looking at faster and more reliable two-wheeled enforcement from the likes of Kawasaki and BMW.

In order to survive, the Factory had to expand their civilian marketing base competing with the Japanese, particularly Honda and to a lesser extent the European bikes.

With the advent of counterbalancers in the year 2000 Softail series, every Big Twin was finally able to suppress vibration. The other Big Twins dealt with their vibration issues earlier on with a different methodology. The touring FLT series also known as Baggers became rubbermounted in 1980 closely followed by the hot rod FXR in 1982 and the introduction of the Dyna models in 1991. I refer to these models as rubbermounts. Rubber mounting the engine, transmission, and rear fork is an external method to dampen vibration. Internal Softail counterbalancers dampen vibration from within.

The traditional Harley-Davidson lineup became totally vibration-suppressed in 2004 with Sportster rubber mounting. For the first time, the hot rod Sportster could also function as

a touring bike. This vibration dampening progression over twenty-four years was a meaningful and progressively successful attempt by Harley-Davidson to make ready for mainstream riders.

However, H-D chose to suppress vibration instead of fixing it.

Fixing the Symptom, Not the Cause

This is an easy to understand methodology when realizing that the vibration cure involves changing the cylinder angles away from the traditional and beloved 45-degrees to 56 or 60-degree configurations like the V-Rod, Japanese Metric models and the 2008 Buell.

Overhead cams and a host of other changes unacceptable

to the Harley rider of the day closely followed cylinder angles for necessary vibration improvements and increased rpm potential.

Suppressing vibration had the huge and beneficial benefit for Harley and the riding public. The bikes became more reliable. On previous unsuppressed vibration models like the Shovelhead and Evo Softails, insidious vibration operating at many frequencies, some not noticed by riders, broke external bracketry. Vibration left coils hanging by their wiring and plug wires; cracked top motor mounts exacerbating the vibration that broke them further; frame brackets and in extreme cases the frames themselves most frequently occurring on the left side beside and underneath the lower rear primary cover.

Vibration also reduces engine and motorcycle parts longevity.

## Why Address Engine Vibration?

Vibration is not only the bane of a comfortable ride and the premature cracking of motorcycle bracketry it runs counter to an efficiently running engine.

Why spend all the money required to make a Harley faster if engine components are not working in perfect harmony? Internal engine discord will rob small amounts of otherwise usable horsepower and torque. Coordinated harmony helps eliminate internal frictional loss of horsepower.

Furthermore, the effect of vibration aggravates engine wear.

This is why it is so important other than improving rider comfort to address treatable causes of vibration in a performanced engine.

## External Causes of Twin Cam Vibration

### Wheels and Tires

Misaligned motorcycle components such as wheels improperly installed are a cause of vibration as well as other maladies.

### Directional Rotation

All modern quality tires have a directional rotation. Tire design is also specifically for front or rear fitment, not both. This is crucial because with braking as an example, the front tire has pushing forces applied while the rear has pulling.

Acceleration applies a pushing forward torquing force to the rear wheel while the front is along for the ride. Hence, one-reason rear tires cost more as there are additional coping and wear factors.

It is difficult for me to understand why I sometimes see bikes with a front tire on the rear or vice versa since the front and rear tread patterns are usually so different. Much more common are tires installed backwards. This is due to plain ignorance or a mechanic asleep at the wheel, so to speak. A Braille raised directional arrow for even the blind to feel is also a visual instruction for the seeing. The arrow direction must be the same as the wheel's rotation on the bike.

Improperly installed tires will wear unevenly causing a distressed ride to the point of chunks of rubber flying off in extreme cases. I remember a now defunct tire manufacturer

that was the flavor of the day a few decades ago. Molded directional arrows on the tire sidewalls were backwards. Small chunks began to fly off the rear tires followed by larger pieces roughly measuring up to five inches square.

The manufacturer quickly corrected the mistake as the complaints flooded in. These tires disappeared as quickly as their introduction because of consumer fears.

At Heavy Duty Cycles, we cut tires in half that we find installed rotationally

backwards. We want to ensure that re-installation of a potentially dangerous tire by a frugal and ignorant rider is not possible.

Tire Inflation

This is a biggie since many if not most tires are inflated incorrectly. Over-inflation or under-inflation will cause rapid wear and adverse handling characteristics including vibration.

Correct inflation for 1999 Baggers is 36 psi (pounds per square inch), using stock OEM Dunlop tires, front, and rear with a single rider. Air pressure for a rider and passenger remains at 36 psi for the front tire and increases to 40 psi for the rear tire inflation.

Never exceed the maximum tire pressure. Molded into the tire sidewall is this maximum psi figure. In fact, find all the pertinent tire information by reading sidewall molded-in information specifications.

Inflate tires when cold. Cold means the tires are at ambient temperature and not been taken up to operating temperatures

by riding any distance.

Air expands with heat.

Fill Performance Tires with Nitrogen

Nitrogen has some qualities that make it an excellent substitute for air. The first is that it neither expands nor contracts with temperature changes. Therefore, tires stay at correct tire pressure, no matter the temperature.

The second advantage is that Nitrogen does not have water as a component as air does. Condensation will not form inside Nitrogen filled tires, rusting out rims and spokes and oxidizing alloy rims.

Air is prone to leakage, especially with mag wheels. Nitrogen, while it can leak is less likely to do so.

The one downside is that air is free while Nitrogen is not.

I carry Nitrogen at Heavy Duty Cycles for customers, as do some car performance shops and medical supply houses. Vehicles like ambulances use Nitrogen as a safety issue. Racecars and motorcycles of all types use Nitrogen for safety and tire pressure consistency reasons.

## Worn Tires

It-goes-without-saying, that badly worn tires are unsafe and if worn unevenly, which is common and therefore is a source for vibration.

## Radial Tire Runout

When installing a new tire, it is a good idea to put the rim on a truing stand to measure radial runout. Check lateral runout at the same time to ensure it is within .030”.

Radial runout is up and down variance from concentric. If the rim has, say .030”or less radial runout proceed with tire mounting. If there is, more than .030” a wire spoke rim needs re-truing to bring radial runout within specifications.

A cast rim may need replacing depending on how far above .030” the radial runout actually is. Harley-Davidson advises replacement if radial runout exceeds which equates to .03125” but it, is easy to say that. Not everyone has deep pockets so compromise may save some money.

I do not get antsy to replace a rim for poorer customers until hitting .040” radial runout. I explain the pros and cons

and let the customer decide. However, if it goes higher than .040”, then it becomes a safety issue in my mind and I will dig in my heels and refuse to re-install the rim.

After tire mounting, check radial runout once again. If up-down runout goes beyond .090” (2.30 mm), then we need to remount the tire again. Tires are not as precise as metal rims. This is why the radial runout measurement climbs from .030” on a rim to .090” on a tire. If the tires radial runout stays at .090” or more then the tire is defective and in need of replacement.

A hopping tire will create much vibration.

Lateral Tire Runout

We now check lateral or side-to-side tire runout.

Since we have already checked the rim for both radial and lateral trueness and know it to be good within .030”, the tire should have a maximum lateral runout of .080” (2.00 mm). If this runout exceeds this amount, then remove and remount the tire. If still above .080”, then replace the tire.

## Wheel Bearings

Wheel bearings must be in serviceable condition or else they may cause adverse handling reactions and vibration.

End spacing is also crucial for the same reasons.

## Wheel Balancing

Balancing is crucial to good handling especially at high

speeds. Unbalanced wheels cause much vibration.

Static balancing is with the wheel at rest, being hand-turned while in a wheel truing stand or jig. A lathe with a deep-enough bed is also excellent for static balancing. The H-D static wheel-truing stand is part number HD-95500-80.

Dynamic or spin balancing is superior especially for higher speed riding. The static balancer costs a hundred or two. The Heavy Duty Cycles dynamic balancer made by Snap-On cost thousands. This is the financial reason many shops do not have a dynamic balancer.

Balanced wheels should be at the equivalent of 60 mph to within a half-ounce weight or about 100 kph (kilometers per hour) at the equivalent weight of 14g (grams).

On H-D cast rims, if requiring a balance weight of 1 oz. (28g), or more in one location, apply half the required weight on one side of the rim and the other half of the weight on the other side of the rim. These cast wheel-balancing weights glue to the alloy with an appropriate bonding agent such as Loctite Superbonder 420.

Wire wheel weights attach in the center around the spoke(s).

The maximum allowable total weight is 3.5 ounces for a Harley FLT wheel.

Look for and resolve another problem a wheel weighting of this magnitude is attempting to mask.

Out of True Rims

Before balancing, it is imperative to ensure the wheel rim is true.

Truing laced wire wheel rims is a very skilled job where art may transcend science. I know a diabetic-induced blind person whose truing skills are legendary in my neck-of-the-woods. He trues rims by tapping the spokes with his spoke-wrench much like a tuning fork. He tightens and loosens to perfection by sound.

Tightening and loosening various different spokes to build a perfectly round and

concentric rim on a spoke wheel ensures trueness. All this, and there are minimum torque specifications that the mechanic must adhere to depending on gauge (diameter), of the spoke.

Check your manual for the torque specifications as they can vary wheel-by-wheel type. One such specification is to tighten each spoke by turning the nipple with a spoke wrench to 40-50 in-lbs (4.5-5.7Nm). The measurement is in inch-pounds not foot-pounds. It is common for new mechanics or those not paying attention to mix the two with disastrous results as in parts breakage.

Tightening a spoke or combination of spokes will pull the rim in their direction. Loosening spokes will have the opposite effect. However, this must be within predefined limits, as all the spokes must have a minimum torque to retain the structural integrity of the wheel under riding stresses. Complicating the controlling of side-to-side warpage by the process just described are concentricity requirements. The rim must be equidistant to the center hub axle throughout its circumference.

If the rim is pulled in too far at say the 9-o'clock and 3-o'clock positions then the 12 and 6-o'clock positions will be too far out to form a perfect circle resulting in wheel "hop" at any speed but certainly aggravated and intensified at higher speeds.

Some of the simplest mechanical jobs can rapidly become complicated, requiring cause-and-effect thinking on different levels at the same time much like Spock playing 3-level chess. In short, truing a spoked wheel is a very skilled job requiring much practice and trial and error.

Mag wheels cannot be trued in the same way and require differing skill sets. Minor dents and warpage may require the judicious use of a mallet usually combined with heat. The mechanic may make special tools the shape of the outer rim

to use as templates in combination with the use of a mallet and heat. Polishing may remove minor cosmetic defects resulting from the original damage and/or the repair.

Damaged wheels such as dented rims, out of round rims and broken spokes all contribute to rider safety concerns as well as vibration.

Then there are the improperly laced spoke wheels that are another whole story.

Steering Head Bearings

Consult your manual for specific bearing pre-loads for your year and model. Pre-loads and swing-by checks are different for every model Harley.

Check steering head bearings for adverse wear such as

scoring, brinelling or pitting.

Brinelling describes surface deformations or metal portions are compressed. Pitting describes the removal of small pieces of metal from the metal surface.

I ask the obvious question. Why inspect except for causal factors that may need correcting? There is so much work before actual inspections, for the cost, why not replace the bearings and mating races with new every time? This is what I do.

Many times both wheel and steering head bearing races are difficult to remove. Do not succumb as many do and only replace a new bearing and leave the worn race in place. You have wasted your time and money if so.

After installing and torquing to specifications the bearings and races and the fork crown nut/bolt, install the forks with wheel and fender. If possible, leave cables removed for best swing-by test results. Raise the front wheel with bike upright so the front fork has free movement. Now perform the swing-by test. For example, an FLT swing-by test should always result in the forks returning to center when moved because of their unique reverse crown setup. Move the front forks to either fork stop. Release the fork. The fork assembly should swing back past center but not all the way to the opposite fork stop. The fork should then return unassisted to the center position. If the fork swings too far, bearing load is loose. If there is binding or the fork does not center, then bearing preload is too tight.

Softails and Dynas have different swing-by tests to ensure correct bearing preload.

Improperly Adjusted Belts

When a Twin Cam rear belt is maladjusted, the rear wheel skews one way or the other depending on which side adjusts too much relative to the other. Consult your manual for the correct belt adjustment procedure for your particular year and model Twin Cam.

As a generality, I will describe the procedure for a 1999 FLT series. Belts do not wear as unevenly as a chain, which can have exaggerated loose and tight spots. However, they can stretch at different rates along their length. Worn belts tight and loose spots causes jerking when a tight spot rounds the drive sprocket thus causing a vibration. As with a chain, always look for loose and tight spots. Adjust belt tension at the tightest spot.

Frictional heat from motorcycle riding will also affect

temporary stretch. Adjust belt tension when cold.

It is imperative to adjust the belt under normal weight operating conditions. Adjust belt tension with the usual rider or one with similar weight seated on the bike.

There is a specific H-D tool for checking tension called a belt tension gauge, part number HD-35381. Place the tension gauge half way along and underneath the upper or lower belt depending on year and model. Many belt guards have a vertical slot about a quarter-inch wide with measurement markings. As the belt deflects to ten pounds, it is easy for the mechanic or owner to see if deflection is correct or if adjustment requires correction. Apply the tension gauge to a pressure of 10 pounds (4.5 kg), and use the axle adjusters, if necessary, to ensure a movement that Harley calls deflection of a minimum to a maximum ⅜", which translates into a minimum of 7.9mm (millimeters), to a maximum of 9.5mm for the metric crowd in most of the rest of the world.

Before adjusting the axle back to the rear if too much belt movement or forward if there is not enough deflection it is necessary to loosen the castle axle nut. This is a specialized nut where one end looks like a castle rampart. In between, the rampart appendages, a large cotter pin or lynch pin, is stuck in a drilled hole through the diameter of the axle and out the other side between opposing ramparts thus preventing the castle nut from turning. After removing the cotter, or lynch pin, as the case may be, loosen the axle nut thus allowing the axle to be drawn back or forward. Turn the two axle adjuster nuts, one on either side of the rear fork an equal number of turns or distance to ensure correct alignment of the wheel. Of course, we always check that they have equidistant turns before the adjustment procedure. Torque the axle castle nut to 60 ft-lbs or the metric equivalent of 81Nm. Re-install the cotter or lynch pin.

The 1999 1 ⅝”-wide belts use Aramid fibers colloquially known as the bulletproof Kevlar. Belts have gradually decreased in width over the years until 2007 when the X7N second-generation carbon fiber 1”-wide belt came into use. The X7N utilizes thicker fabric for increased tooth nub strength and durability. The polyurethane outer core body opposite the nubs is the same material used in the 1999 Aramid fiber belt. The tensile cord in the middle is comprised of the new carbon fiber

I mention all this to emphasize consulting the H-D workshop manual before making adjustments. As the reader can see from the following chart that belt deflections have changed, differ between models and are done without rider weight. The upgraded deflection tool, part number HD-35381A is also different.

Comparative Belt Tension Specifications

1999 (1⅝”-wide 136-tooth USA) Rear Belt Tension Specification

2002 (1⅛”-wide 135-tooth USA) Softail

2005 (1⅛”-wide 133-tooth USA) Dyna

2007 to Present, 1”-wide Rear Belt Tension Specifications

There have been changes to tensions (belt deflection), and procedures to obtain correct tensions because of the new type of belt used. The 2007 and later Twin Cam Harley should be at ambient temperature with the transmission in neutral. As always, check in the belt's tightest spot. Check tension at 10 lbs (4.5 kg), pressure midpoint at the bottom of the lower belt. H-D tool (HD-35381A) will assist the inexperienced.

2009 140 -tooth Rear Belt Tension Specifications

Loose Rear Fork Bolts

Loose Softail rear fork bolts or as in the case of

rubbermounts, loose rear fork pivot-shaft-nut(s) will also cause vibration because the whole rear end will be able to shift and/or twist particularly while cornering.

Consulting the manual for year and model Twin Cam will allow correct torque application in specified intervals.

The 1999 Twin Cam FLT Baggers, for instance require a torque check initially at 1000 miles (1600 km), and every 5000 miles (8000 km), thereafter. The torque specification procedure for these models is to hold the right side nut on the pivot shaft with a socket while tightening the left side locknut to between 40 and 45 ft-lbs. The metric measurement is 54 to 61Nm.

Broken Frames

Early FLT frames may break under the seat after rusting out.

I have never seen a Dyna or FXR frame break.

Rigid mount frames may also break underneath and to the rear of the primary chain case.

Sometimes vibration causes the frame cracking or breaking but once occurring always results in extreme and dangerous vibration.

Fork Oil

All Twin Cams with the exception of the Springer Softail have hydraulic front forks.

The only function of the internal fork springs is to hold the bike up.

A physical property of air is that it will compress. Therefore, air inside the fork will compress.

A physical property of oil is that it cannot compress. Oil's function is to transfer through orifices in the dampening rods below the springs to regulate compression and extension of the forks. The oil viscosity or its rate of flow, orifice hole sizing, and number of orifices combine to produce dampening rates for different functional types of bikes. Inadequate oil in the forks screws all this up. Lack of oil in one slider with adequate oil in the other is a recipe for an accident.

Did I leave out vibration?

Engine Tune

Incorrect ignition timing as part of a poorly tuned engine can easily cause rough running and vibration. Too far advanced ignition timing due to a faulty Ignition Control Module (ICM, ECU or ECM), sensor inputs can cause rough running, engine damage, and vibration with modern Twin Cam engine management systems. This malady can be caused by defective MAP (manifold absolute pressure), CKP (crankshaft position), or in early Twin Cam models, the CMP (camshaft position) sensors.

Primary Chain

The primary chain resultant vibration will be a symptom of my examination of the sprocket shaft flywheel scissoring out of true. Think about it. If the left flywheel scissors, the integral sprocket shaft will also go out of whack. Instead of

running true, it will have a radiused curvature moving forward, down, back and up. This will snap the primary chain too taut, then too loose, and them back and forth causing wicked vibration.

There are three primary (forgive the pun), causes of engine to transmission connecting chain-induced vibration. More correctly, the primary chain is not the initial causal factor but a resultant one passing the grief down the line as it reacts to other improper conditions. The next two are obvious after who would think, Twin Cam flywheel scissoring, when pointed out, misalignment and wear.

Wear occurs naturally with time and usage. Chains wear unevenly and when worn will have tight and loose spots. A badly worn primary chain's tight and loose spots will cause a whipping or jerking action. Insufficient lubrication or extreme

wear will cause links to kink and not fully extend.

Chain misalignment causes much grief as the chain grates against the inner or outer sprocket as the case may be cutting its new path causing vibration and a plethora a metal fragments to wreak their own havoc. For example if the front sprocket is further outwards from the clutch hub sprocket, the chain will recut a path on the outside of the engine sprocket and the inside of the clutch hub sprocket until the chain has a straight pathway. If the engine sprocket location is inwards of the clutch hub sprocket position then the opposite cutting action will occur on the inside of the engine sprocket and the outside of the clutch hub sprocket.

The primary adjustment shoe not aligning with the chain causes you guessed it, premature wear, and sooner-than-later vibrational destruction may compound all of this.

## Alignment-Wheel Trueness: 2008 and Earlier

To ensure rear wheel alignment with the motorcycle, in this case an early FLT model, the first step is to make certain that the wheel is true.

The FLT mag wheel has a cast rim. By the way, the term mag wheel is Harlyese vernacular that refers to a solid wheel. Harley cast wheels are not true mag or magnesium wheels.

## Mag Wheeel Truing

Determine runout by placing the wheel in a wheel-truing stand. Check for lateral and radial wheel movement via a

stationary attached rod that indicates any untoward runout immediately after the wheel is hand-spun. I prefer using a magnetic based micrometer called a dial indicator that measures in thousandths-of-an-inch. Lateral or side runout must not exceed or the equivalent .0469 thousandths-of-an-inch or in metric, 1.19mm. Radial or up and down rubout should not be more than or the equivalent .0313 thousandths-of-an-inch or .79mm in metric.

If exceeding either measurement, the Factory advises to replace the rim. In practice, we at Heavy Duty Cycles can do minor re-truing procedures with heat, custom-made tool inserts to protect the rim integrity, a mallet and a hydraulic press.

Mag wheel offsets are not important to measure for wheel trueness since any offset is fixed and thus non-adjustable. Offset means that the rim is set to one side of the center wheel hub. This is common with various H-D wheels. Offsets are

perfectly safe ensuring correct fitment with surrounding parts and of course, wheel alignment with the model in question.

## Spoke Wheel Truing

Truing wire or spoked wheels, first requires measuring the prescribed offset. Simply, lay a straightedge across the flat part of the center wheel hub, usually the disc brake side. The perfectly flat straightedge extends past the outer wheel rim. Measure the distance with a Vernier caliper, which is an extendable ruler that measures in gradients down to a thousandths-of-an-inch from the straightedge to the rim. Sample offset measurements for a front spoke wheel might be 1.565” (39.8mm), and for a rear, spoked wheel might be 1.722” (43.7mm). Consult, the Harley-Davidson manual for particular year, model and type of wheel offset specifications.

Adjust offset by loosening the spokes on one side of a rim and tightening them on the other side or vice versa, depending on whether an increase or decrease is necessary. Next, perform lateral trueness procedures, followed by radial trueness. Finally, then tightening each spoke nipple to the specified torque in in-lbs short form for

Alignment-Stabilizer Links: 1980-2008

The entire powertrain is rubbermounted and isolated from the frame, thus shielding the rider from real-world vibration. The engine, primary system, transmission, rear fork and rear wheel are rubber isolated. Keeping these components as well as the front wheel in alignment in order to stop further vibration as well as containing existing powertrain vibration is the name of the game for a perfectly functional motorcycle riding experience.

In 2009, H-D further improves this system by isolationg the engine on four corners instead of a 3-corner triangle configuration. Procedures are much the same but consult the H-D manual for specific differences.

The following alignment procedures depend on the first step, which is ensuring the correct rear wheel position. Determine if measurements from the center of the rear-fork-pivot-shaft to the center of the rear axle on both sides of the motorcycle are equidistant. The manual instructs on how to make a simple gauge tool to accomplish alignment measurements. If one side's measurement is greater or less than the other opposing side's measurement, then adjust the axle adjusters until attaining equal measurements.

Raise the motorcycle on a stand or center hydraulic lift. It is only necessary that the rear of the bike to rise off the

ground. Remove the top motor mount stabilizer-link attachment. This is on the left side of the bike, near the top of the engine between cylinders and underneath the gas tank. Next, loosen the lower front engine rubber-mount bracket bolt after taking the voltage regulator off. It is not necessary to disconnect any wiring. There will not be any damage leaving the regulator hanging out of the way by its wiring. Even so, I always support the regulator. Also, loosen the two front bolts holding the engine mount outer plate to the frame.

The next step is to confirm that the front and rear wheels align. Place straightedges along either side of the rear wheel that is long enough to extend to the front wheel. Alignment of the front wheel to the rear wheel is readily apparent if it centers between the two straightedges, one on either side.

At Heavy Duty Cycles, we use sliding door tract from the

local home renovation store. This tract must be straight for doors to move inside them. We also check their integrity with straight edges.

A Mistake at the Beginning Extrapolates Onwards

Now, it is easy see the reason that the first step of rear wheel alignment in the rear fork is so crucial to the rest of the alignment procedure. A mistake at the beginning continues throughout the motorcycle.

To center a misaligned front wheel, use a tool called an Open End Crowfoot, Snap-On part number FCO18, to loosen the two nuts on the bottom stabilizer link. Now adjust this stabilizer to correct any misalignment.

Stabilizer links are dual adjustable with a threaded eyelet

at either end, one a right hand thread with the other a left hand thread. A Heim joint is the correct nomenclature for each eyelet and its housing appendage. A thin hard rubber cushion surrounds the eyelet in the end of the Heim joint opposite its female threaded portion. The stabilizer with an eyelet in each Heim joint at either end of the adjuster bar accepts a bolt to attach the engine to the frame via a rubber mount. Turning the right and left hand threads out extends the length of the stabilizer and turning them in shortens it. The center adjusting stud with right hand male threads on one end and male left hand threads at the other attaches and connects the Heim joints. The adjuster bar has an integral nut that is fixed and part of it. This nut turns to extend the stabilizer longer or minimize its length with the Heim joints and their bolts still attached to the frame and engine. One Heim joint can be at an angle to the other say, at 90 degrees to the other or any other angle for that matter. A lock nut on either end of the threaded adjuster shaft locks the Heim joint in place as well as keeping the stabilizer length fixed until the

next necessary adjustment.

Other configuration stabilizer links are available from the aftermarket including a deluxe hydraulic unit between the Heim joints to absorb vibration.

Adjusting the bottom-front engine-stabilizer will allow centering of the front wheel between the straightedges by aligning the rear wheel to the front. Horizontal misalignment adjusts via the bottom stabilizer link. Vertical misalignment adjusts via the top stabilizer link.

The adjustment rubber-alignment procedure is as follows. Tighten the two front engine mount bolts to between 33 ft-lbs and 38 ft-lbs or in metric, 45-52Nm (1999 FLT), after centering, the outer plate so that the rubber centers and equally compresses side to side. After taking the front engine-

mounting bracket off its bolt, push this bracket to the front or rear until the rubber compresses equally forwards and backwards. Now, the front engine-mounting bracket installs (35-45 ft-lbs/47-61Nm), to the centered rubber mount. Recheck alignment to make sure tightening everything down did not throw anything out of whack. Re-install the voltage regulator and lower the rear wheel or the bike, depending on how elevated, to the ground.

Stabilizer links and rubbers wear, as they do lots of work since they are the alignment and vibration controllers. Check stabilizers for wear every 10,000 miles (16,000 km). I prefer checking them more frequently for the following conditions; torque, alignment, worn eyelet holes, which usually egg-shapes, seizure, bound isolator mounts and powertrain to frame contact.

2009 Engine Isolation-mounting System

Handling and vibration control improves once again by expanding the previous 1980 to 2008 three-mount-system to four.

The single center upfront isolation mount replaces with two side-mounted vertical isolation supports. One mounts at the base of each frame downtube and tie together with a single link. Now each of the four corners the engine have isolation-supports instead of a shaky triangle.

The new supports are supposed to last the life of the motorcycle. This is great but I must say that I have heard this type of claim before notably for the early disastrous 1999 to 2006 Twin Cam spring-loaded timing chain tensioners.

Logically, the new isolation mounting system will last much longer since there is extra support that does not allow

wide swings in engine vibration especially at idle. The improvement effect is most noticeable at idle. The engine does not shake as it has from 1980 right up until 2008.

Air Shocks: 1980-2008

Air shocks need checking for correct air pressure. We also want to ensure one or both sides do not have an air leak. In general, air fill is from one fitting with connecting air-hose tubes.

The right saddlebag may contain an air-pressure-specification information decal Location may be elsewhere depending on model and year.

Motorcycle load is crucial. Using the 1999 Bagger as an example, maximum air pressure is 35 lbs. However, the

pressure may be set according to comfort preference from 0-10 lbs if rider weight is 150 lbs or less. With each additional rider weight of 25 lbs, add 1 lb air pressure. Each 50 lbs of passenger weight requires an addition 1.5 lbs pressure. Luggage weight in increments of 10 lbs necessitates an additional 3 lbs air pressure.

## 2009 Shocks

There are now dual adjustable air shocks replacing dual non-adjustable air shocks.

## Shock Oil

It is crucial to ensure both air shocks are oil-filled. Store upright when removed so oil cannot drain. It is not possible to fill many air shocks with oil once depleted via improper storage or leaking.

One shock with oil and the other without will create a hazardous ride beyond the ensuing vibration.

Front Fork Air Suspension

Twin Cams equipped with air front forks should have a maximum 25 lbs air pressure (1999 Baggers), with riders under 150 lbs having a comfort preference between 0 and 15 lbs. With each additional rider weight of 25 lbs, add 1 lb air pressure. Passenger weight is immaterial in the case of front fork-air-pressure, as this weight does not influence air pressure requirements. However, luggage weight in increments of 10 lbs necessitates an additional 1 lb air pressure.

This does not make sense to me since luggage storage is

behind the passenger and logic suggests should not have any effect.

I cover fork oil elsewhere.

## Vibration Specific to Counterbalanced Softails

The upper rigid motor mount to the top frame member has been a notorious source of vibration for more than a century on rigid engine mounted Harley-Davidson's. Hold down attachment bolt(s), stud(s), nut(s), or a combination thereof, as the case may be, first loosen and may drop out entirely onto the ground or between the engine and inner primary cover.

Mild vibration with loosening of attachment fasteners to severe shaking ensues, setting the stage for more hardware undoing and subsequent parts and component cracking and

finally breaking.

It is crucial with rigid mounted engines and transmissions not to stress parts during assembly as vibration will eventually crack and break them. The top motor mount is a prime candidate for breakage in these circumstances closely followed by the two front external primary-to-engine hold-down bolts. Even those two thick and strong aluminum cast engine hold-down lugs to the frame will snap when stressed.

## The Fault Lies with the Last Assembler

This fault lies entirely with the last assembler since any competent mechanic knows to shim these and many other parts like two-piece fatbob gas tanks if mating mounting brackets do not align correctly and precisely so that when attachments are tightened and torqued down there will be

absolutely no twisting or bending stress to force the parts together.

Engine front lugs do not always mate perfectly with their frame counterpart. I do not know how many times, I have heard wrenches blame aftermarket frames in particular and parts in general for lack of precise fitment. They bemoan, quite accurately in some cases the lack of quality control.

Just as quickly, they will praise the use of OEM (Original Manufacturer Equipment), parts as they assume wrongly that they always fit correctly. However, the same lack of exact fitment situation is very true and common with stock H-D frames and their motor mounts not aligning precisely with Harley engine mounting lugs.

The subsequent danger is that these wrenches will not check OEM fitment for possible stress misalignment and set

the stage for vibration and parts damage.

## Engine, Transmission and Primary Chassis Alignment

First, inspect all mating parts like the underside of the engine front hold down lugs and the attaching companion frame lugs. The surfaces must be smooth without raised casting defects. They must be level and machined flat.

Place the engine in its frame mounts and observe if any rocking motion will ensue when jiggling the engine. If so, machine the offending high spot(s) flat. If this is not possible, then shim the engine mounts to eliminate any rocking.

Then mount the transmission. Align the inner primary with engine and transmission. It is crucial that it line up with no stress whatsoever on the front mounting tabs as they easily

crack and break under even slight twisting pressure. Sequence torque all transmission and engine hold down bolts to specification.

If there are loose engine and/or transmission hold down bolts mounting the front of the engine to the frame (2), rear engine to transmission bolts (4), bottom transmission to frame (2 or 4) and the frame center down tube to transmission top will all cause much vibration and sooner than later parts cracking and subsequent breakage.

## The Unique Softail Frame

The 2000 to present Softails are unique to Harley-Davidson Big Twins by incorporating the strength and rigidity provided by an attached engine and transmission as an effective frame member. This is not unique to many other motorcycles where powertrain components have dual functions, one of which is a

supporting frame member.

The term for this type of frame is a backbone frame.

A cradle frame surrounds the engine and transmission. All traditional Harley frames are of the cradle variety except for the new Twin Cam Softails.

However, the 2000 to present Softail is neither a pure backbone nor a cradle frame but a hybrid of both.

Out of Time Counterbalancers

The Softail engine counterbalancers will wreak vibratory havoc if out of time with each other.

The easiest way for the counterbalancer to be out of time is improper assembly by not aligning the timing marks correctly.

Bearing failure can precipitate an out-of-time situation but is an extreme case; always replace all four counterbalancer bearings whenever disassembling the engine. There are two bearings front and rear for a total of four with one in the crankcase (OEM 8989), and another in the housing (OEM 8991).

The only other way for an out-of-time situation to occur is extreme wear. This may include the hydraulic adjusters collapsing; excessive sprocket and chain link wear or the chain missing a roller bushing(s). The chain would have to jump a sprocket tooth or teeth to bring the assembly out of time.

Always replace chain and sprockets as a set during a bottom end rebuild. This is especially important with the counterbalancer mating parts, as there is much labor involved to get to them if requiring service.

With that said, the following also applies to rear chains and sprockets as well as belts and their respective pulleys. It is not a good practice to replace only one pulley, sprocket belt, or chain as the case may be. If one part is worn enough to replace, then it only follows that the other two need replacement as well. A badly worn pulley will have worn the belt as well, which of course, will have worn the other pulley too.

Chains are even more demonstrative in their wear capabilities of mating parts. Sprocket teeth wear in the opposite direction of chain rotation. First, the teeth will

narrow down from the top narrowest portion of the tooth. The narrowed top will then start to wear directionally as if pulled backwards. This hooking will increase until the tip breaks right off. The chain will begin cutting a new tooth, albeit a shorter one and the process will start all over again. This will continue until the chain will start skipping over the progressively diminutive teeth.

The same will be happening to the other sprocket(s). The workshop manual says to replace sprockets if they show severe hooking. In my opinion, it is just too hard to get at them to allow any wear without replacement.

At this point, the chain will also be badly in need of replacement, as it will be stretched very unevenly, rollers will be damaged, kinking may occur until a link finally breaks, snapping the chain. We do not want any of the above to begin happening in an enclosed engine casing as Armageddon may ensue.

Always inspect the chain for missing roller bushings, side plates, and adverse wear from turning, which may include burn marks. Once again, my credo is to replace, replace, replace when a wearing part is deep inside an engine or transmission.

However, granting wear leeway to easily accessible parts is a no-brainer. Monitoring wear visually, by feel or by sound without difficulty is the key since replacing a surface part does not incur extra labor.

Counterbalancer and other engine bottom-end parts are definitely not in this category.

Bluing: An Indicator of Heat Wear or Heat Treating

Bluing is usually an indicator of heat wear. In the case of a chain, bluing indicates severe compromising of the integrity of the chain.

Other bluing may be part of a manufacturing heat-treating hardening process. Connecting rods bluing around the big ends, which rotate on the bottom-end bearings surrounding the crankpin are a case in point.

Bluing on Shovelhead or Evolution connecting rod big ends means that their replacement is necessary as well as diagnosing and fixing the cause of the extreme heat so the bluing will not repeat. These rods have separate hardened races pressed into their big ends.

However, on the Twin Cam, the connecting rod big end

bearing races are not separate but made from parent material. The bearing surface is integral with the rod's big end. This area is heat treated to give the parent bearing the necessary hardness to withstand the rigors imposed by the bottom end. Bluing on Twin Cam con rod big ends is normal and does not indicate a heat or extreme wear problem.

Improperly assembled sprockets may also show wear on their inside edges around the balance shaft necessitating replacement. Therefore, it is crucial to replace all mating parts even if wear is not evident on them all. This includes all three sprockets and the chain as well as hydraulic adjusters if they fail leak down tests.

The Third Counterbalancer Sprocket

The third sprocket; what is that all about? Well there is a sprocket at each counterbalancer end and the third is a center

sprocket on the flywheel pinion shaft.

Sprocket Alignment

There is a procedure for checking alignment of the flywheel pinion-shaft center sprocket to the balance-shaft. The balance shaft sprocket alignment tool (HD-44064) slides over the pinion shaft up against the center sprocket face sometimes called the timing chain gear.

A simple change of words for the same part sure can cause a lot of confusion!

The alignment tool's appendage can now swing over the front and rear counterbalancer sprocket-face for alignment measurement purposes. Alignment must be within .014" (.356mm). Adjust the alignment if beyond specifications by

replacing the spacer behind to offending sprocket with a correct dimension one. The spacers come in nine sizes in .010” (.054mm) gradients from .130” (3.302mm), thick, (OEM 14780-00), up to .210” or 5.334mm, thick (OEM 14788-00). The part numbers increase by one digit preceding the hyphen per size; 14781-00 is for a .140” (3.356mm), thickness; 14782-00 is for .150” (3.810mm) thickness and on it goes up to part number 14788-00.

Timing the Counterbalancers and Chain

Harley-Davidson enables this crucial and otherwise complicated timing procedure very simply. The Factory’s addition of placement indicators, timing marks and color-coding achieves this. Without these handy assists, the timing procedure would get very complicated, very fast.

First, the following timing marks on the sprockets and

timing link colors on the counterbalancer chain must face outwards. Duh, how else would one see them? If put on backwards, one would quickly discern that, they have to face outwards for timing.

Front and rear sprockets are stamped F and R for correct placement within the chain.

With installation, always use new sprocket nuts and locks. If new fasteners are unavailable, once again, red Loctite becomes your best friend when reusing old nuts.

With the F and R stampings at the 12-o'clock position within the chain, a timing mark will be at the 3-o'clock position on the front F sprocket, while a similar timing mark will be at the 9-o'clock position on the rear R sprocket. The center-sprocket-timing-mark should be at the 6-o'clock position on the pinion shaft. This involves rotating the

flywheel assembly so that the sprocket presses on with its timing mark directly below the crankpin oil-feed hole in the pinion shaft.

There are three colored chain links for timing purposes. The middle colored link in the chain must seat directly below the center-sprocket timing mark. The fore and aft colored links will now align with their mating sprocket timing marks.

Retaining Pins: Balance Shafts

Counterbalancer shaft housings are also marked F and R for correct placement in the front or rear engine-casing compartments.

The balance shafts may be used front or rear but install with the smooth side down.

Now insert the retention pins for the balance shafts (HD-44062), inside the supports, fore and aft thereby locking the balance shafts in time with the engine. For timing to be accurate, I will repeat that the center-sprocket timing mark must align directly below the pinion oil-feed to the crank. The center of the otherwise round pinion shaft flat for oil pump alignment is directly above the center-sprocket timing mark and below the crank oil-feed hole.

It is essential that the retention/retaining pin in the end of the shaft do not become loose or dislodged; Armageddon will once again threaten.

Hydraulic Tensioner Leak-Down Test

A simple manual leak down test similar to those used on Shovelhead lifters is a crude yet effective quick way to

determine if there is too much wear for the hydraulic action of the counterbalancer chain adjusters to work correctly.

However, we should be more precise since these parts are not easily accessible.

Oil will not compress like air and some other fluids. This is the physical principle of why oil has use in most hydraulic applications. However, if oil escapes past worn or faulty hydraulic components then the hydraulic action will be partially and in extreme cases totally lost.

Varying degrees of vibration from mild to wild accompanies an improperly tensioned counterbalancer chain. There is not quite any noise as loud, destructive or rider panic inducing as a flopping counterbalancer chain inside a Softail engine's right side.

There are two counterbalancer hydraulic tensioners, one before, and the other aft of the center sprocket. The tensioner comprises of three parts; a small cylindrical hardened steel piston, a tensioning spring, and a plastic vent cap. These parts sit in their guide or bore. The tiny spring's function is much like a much larger front-end fork spring.

The fork spring keeps a bike level while the fork hydraulics does the dampening work.

The hydraulic tensioner spring keeps the piston up holding the counterbalancer chain taut while at rest. The hydraulic tensioning action actuated by oil flow when the engine is running provides the dampening to tension the chain while at work.

There is an interconnect passage in the counterbalancer chain guide. Apply compressed air through this passage while holding your other hand alternately over the front and rear hydraulic pistons. There is an indication of a wear or alignment problem if more air travels past one than the other does.

Disassemble the tensioners and inspect. First, check that the plastic vent cap is on the spring so it can seat under the hydraulic piston. Then check spring free length. It must be at least two inches (2.00”), or the equivalent metric 50.8 millimeters. Now inspect the piston where it goes up and down in its bore. Some polishing is normal and okay but pitting is not.

Again, I ask the rhetorical question, why not replace these parts anyway? I do.

The tensioner guides must not have any grooves or markings deeper than .090” (2.29mm). Again, it is too far into the motor for me to tolerate much wear if any at all.

The chain-guide brackets only serviceable parts are the rubber interconnect, the hydraulic piston (2), the springs, and the 2 plastic vent caps. With leak down failure, after serviceable parts inspection the only option is to replace the rubber-interconnect and redo the test. If this fails, replace the whole chain-bracket guide assembly.

The sophisticated and most accurate way to do a leak down test is to use a cylinder leak down tester (HD-35667A), and tester nipple with 35 psi compressed air into the chain-guide oil passage in the left center timing chest of the right engine casing.

Watch and record pressure changes while covering and

removing your thumb from the adaptor outlet.

Consult your manual for the acceptable leak rates depending on the meter used on the tester. If unacceptable, follow the above procedure for replacing the rubber interconnect to determine how many parts need replacement.

## Chain Guide Screen

A chain guide filtering screen and O-ring fits into the chain-guide oil passage; this screen device filters oil going to the hydraulic tensioners. This is very necessary as small particles of debris can cause tensioners to bleed. Oil feeds into the screen in the chain guide passage from the oil pump channeled by the cam support plate that butts against this passage.

The O-ring seals the juncture.

Rubbermount models have this oil juncture blocked because there are no counterbalancers to hydraulically tension and of course lubricate.

This screen looks identical to the tappet oil screen used with late Panheads, Shovelheads, and Evos located between the external oil pump and the rear tappet-guide-block accessible via a plug on top of the right engine case. I mention this because the tappet oil screen was readily available for cleaning by removing a threaded in plug. There is no such luck with the Twin Cam Softail. What happens if the small screen plugs with debris preventing oil supply to the counterbalancer.

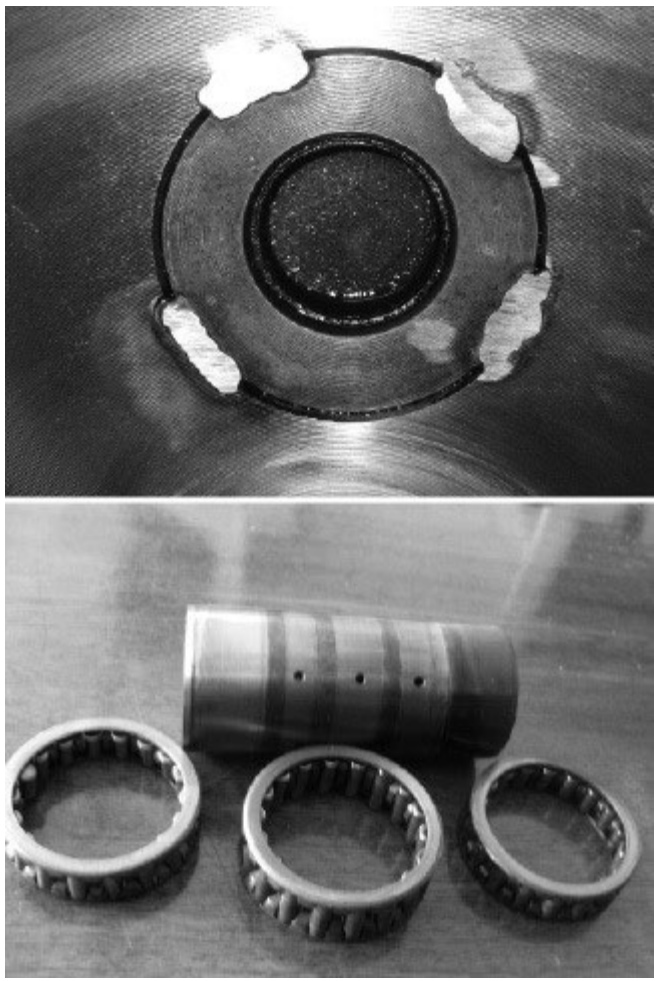
Here we go again…

With that said, I have yet to see a plugged screen during their nine years of current existence.

Shovelhead screens plug all the time. This is another indication of how fine and good Twin Cam oil filtering into the engine really is, compared to the past.

Shovel return-oil camelhair filters were good at stopping small rocks; Evos filtered a high percentage of debris greater than 30-microns (30-millionths of a meter), a huge improvement while early Twin Cams reduced this to 10-microns and in 2006 to an amazing 5-microns.

Welding Crank Pin



Pictured on top is an outer face of a 4 ⅜” Twin Cam flywheel half. The TC96, TC103, and the CVO 110 use this stroke. TC88 and TC95 cranks use a 4.000” stroke. The straight, .007” interference fit, crank pin end that presses the flywheel halves together is the bigger circle in the picture center. The smaller circle is the crank pin end cap. The Twin Cam crank pin is not solid steel but is in reality a hollow tube. End caps press into each end to create a sealed internal oil reservoir to aid in cooling. Oil feeds through the pinion shaft via the oil pump and cam support plate. The oil directs from the center pinion shaft into the right-side flywheel and further out to the offset crank pin. When the reservoir is full, oil forces out via oil pump pressure through three oiling holes shown in bottom picture, to lubricate the three sets of bearings. Mind you, it is a very strong and supportive hollow tube. The hollow center has a diameter of while the solid part is thick, + + for a total diameter of 1 ⅝”. This new-type crank pin is still very much

stronger than what went before. This crankpin shows the baked on oil stain marks between the bearing sets rotational shiny bare-steel paths. Either end of the crank pin that presses into each flywheel half is a different color. They should both be the same. The darker end may have had oil between pin and flywheel when pressed together that baked on. The aforementioned flywheel half would be a prime candidate for scissoring. The captured three sets of eighteen needle rollers in copper-coated cages are supportive, which help break-in procedures, durability, and offer easy wear-spot identification. There are three more bearings than the Evo and late Shovels, and although the same diameter, they are longer. Previous crank pins from 1999 back are solid, much thinner at 1 ¼", have tapers near either end just before threaded portions that allow big nuts to attach the flywheels together. Harley-Davidson repeatedly cautions against any type of flywheel assembly work. The Factory wants mechanics to replace flywheel assemblies when out of true or in need of any type of rebuild. In fact, to reinforce this credo, H-D does not supply

any flywheel components whatsoever. Neither does the aftermarket at this time of writing but surely they will because there are many people like me who will develop rebuild techniques. S&S Cycles, Inc. supplies a straight interference fit crank pin for their versions of the Twin Cam crank. When there is relatively, minor situations that need correcting not everyone can afford to replace the heart of an engine. At Heavy Duty Cycles, we re-true flywheel assemblies that scissor, pinch inwards, spread outwards and those that slant left and right. Flywheel scissoring occurs where one flywheel rotates on its crank pin end front or rear of the other flywheel. This occurs because the interference press fit compromises and allows flywheel movement aided by the progressive increase in torque generated by the TC96, TC103, and CVO 110 engines. The truing procedure may further compromise the press fit so we stainless steel Tig spot-weld the crank pin outer face to the flywheel half. The particular flywheel assembly blowup of the spot welds in the shown picture are ground a little afterwards to remove high spots but

never confronting the integrity of the weld. Theoretically, this welding should only be necessary with the sprocket shaft side, which is the left hand side or more appropriately referred to as the power side. We are too far into the engine not to also weld the pinion side crank pin to right flywheel half as a cautionary move. Truing Twin Cam cranks and welding them afterwards is specialized work that is not difficult for seasoned and proficient mechanics but requires much skill, nonetheless.

## Internal Cause of Vibration: Flywheel Runout

I are going to give some needed background information about Twin Cam flywheel assemblies also known as the crank.

How do they differ from all previous H-D assemblies? Do the differences affect the truing procedure to reduce flywheel and shaft runout?

I will then discuss flywheel runout caused by five differing out-of-true situations; scissoring, flywheel halves too far apart opposite the crankpin, flywheel halves too close together opposite the crankpin and flywheels angled together either way in relation to the crankpin.

Finally, I will discuss correcting the above out-of-true conditions to bring runout at the rims and on the pinion and sprocket shafts within .001” or less.

Harley-Davidson has become very liberal in loosening up their runout specifications as the years progressed after the Twin Cam’s introduction in 1999. They feel this is due to a combination of suppressed vibration where engine rubber mounting and counterbalancing as the case may be masks the pulsing and shaking effects of flywheel runout. The Factory now allows much more leeway in this area because some

opine that the rider does not feel vibration until runout exceeds, can you believe it, .020”! This is twenty times the maximum specification that I and anyone else who has been rebuilding V-Twins for any length of time adhere to, which is .001”.

Perhaps another reason has to do with the exceptional ramping up of production of Harley’s over the last fifteen years, year over year. This extra specification leeway allows for easier and faster production of the crank.

Flywheels

The flywheels press together ever since the advent of the Twin Cam in 1999 on the crank pin.

The sprocket shaft (power output), and the heavier left side flywheel are cast together as is the pinion shaft and the

right side (pinion), flywheel. This innovation developed in the Evolution years and is an important one in combating vibration.

Previous engines had separate shafts from their mating flywheels. Each shaft had a tapered male end that fit into mating female taper in their mating flywheel.

An interference press fit was not the method employed to hold the shafts in place. Their threaded ends extended through the flywheel after the taper for a nut to affix them to the flywheel. In most cases, a small screw will thread into the flywheel inner face to lock the abutting the shaft nut in place.

This worked but not always well. Sometimes the juxtaposed lock screw wound up at the edge of one of the nut flats in the direction of turn for loosening. There is no leeway

in this union of mating parts for the nut to loosen one iota. Other times juxtaposition of the lock screw would position in the center of a flat. In the right circumstances such as crank vibration, the nut could loosen a tiny bit before the lock screw prevented it from moving further. Over time, the nut will move forward and back within its confines until possibly shearing the small locking screw. I have seen this many times over the years. The eventual result is the shaft coming loose in its flywheel with disastrous consequences. Increasing, progressively excessive vibration is the harbinger of this condition.

The shafts are mostly keyed into their respective flywheel centers with a Woodruff key arrangement; the half moon portion of the key fit tightly into a slot cut into the tapered shaft while the flat portion of the key slid tightly into its respective slot in the flywheel taper. The exception was the 1972 and later sprocket shaft whose increased size male taper slid into the correspondingly bigger flywheel female taper,

held by the biggest nut since 1903.

This keying is for two reasons; one it locks the shaft in place to withstand sudden torque demands of a rapidly twisting throttle and two; many times, especially with the pinion shaft, the keying aligns the all-important oil feed passage to the bottom end bearings.

The H-D workshop manuals did not discuss truing the respective shafts to their flywheel. Logic dictates that it is impossible to true a flywheel assembly within a .001" specification at either the rims or shafts if the pinion and sprocket shafts are not at a perfect angle to their flywheels. The old manuals do say that if flywheel trueness is unachievable to "check for a cracked flywheel, damaged or tapered hole, or a sprocket or pinion shaft worn out of round at surface…" 9

All of these conditions occurred in the day. An experienced mechanic would generally spot them while doing a thorough inspection before and during inspection. As always, the thinking mechanic does the best job with the least problems compared with the know-it-alls and the daydreamers.

With the advent of integral shafts in the immediate decade preceding the Twin Cam's introduction, these problems ceased. CNC robotic milling machines machined every shaft and flywheel-casting blank to a perfect true alignment.

This change alone laid the groundwork for a much easier truing procedure, which potentially ended up with truer flywheel assemblies reducing vibration to an all-time low. Well sad to say, I feel that masking vibration by isolating the rider also allowed the Factory to go the other way and allow

looser truing specifications.

## Twin Cam Design Changes

All 2007 and later Big Twin sprocket shafts follow the 2006 Dyna design change in multi-splining the end where the compensator and alternator rotor package slides onto the sprocket shaft with its mating 24-multi-splines. This replaces the fewer ten-⅛" wide splines used on all Harley-Davidson's from circa 1955 to 2006. As it stands, this means that the 2006 Dyna and 2007 and later Big Twin 34-tooth compensator engine-sprocket components cannot retrofit to earlier Big Twins. Nor can the earlier pre-2007 compensator or alternator mechanism fit with the new splined sprocket shaft.

## Alternator-Compensator Rotor

There are other big changes to the new 34-tooth compensator-engine sprocket parts. A compensator that has always been a part of the engine sprocket originally included a big coil spring on the pre-1970 Big Twin absorbs energy shock from the engine back through the powertrain. Since 1970, the spring portion of the compensator consists of a series of fixed and thick diaphragm spring plates.

The new 2006 Dyna and 2007 to present Big Twin 6-speed models have a 4-spring pack for both domestic and H-D International models (HDI).

The compensator portion was a part unto itself that fit between other mating parts in front of the alternator adjacent to the outside of the engine casing. Now the rotor portion of the alternator combines with the compensator as one part that is finely multi-splined (24-splines), compared with

predecessor models (10-splines, ⅛”-wide), in the middle to fit over mating fine splines on the engine output sprocket shaft.

The welded together compensator-rotor unit rotates with the engine with the rotor spinning over the fixed stator, the other half of an alternator to produce current to charge the battery.

## The Twin Cam Flywheel (Crank) Assembly

The first noticeable difference between The Twin Cam flywheel assembly and all that goes before 1999 is the sheer size of many of the component parts beginning with the supporting right side pinion bearing.

It is huge.

The 1999 massive roller bearing is about 3 ½” in diameter;

it is about double the size of that of the relatively skinny needle roller bearing used in the preceding Evo engines. Its function is to support the beefy pinion shaft forged as part of the right-side flywheel.

This pinion shaft was just shy of a ½"-thicker (.400") than its Evo counterpart where it fits through the pinion bearing. A few thousandths of an inch is a long way inside an engine, so these sizing changes are big stuff. This pinion shaft is not going to break, an occurrence while uncommon in Evolutions does happen.

Total flywheel weight is 36 lbs compared to the Evolution's 32 lbs and S&S Evo assemblies at 25 lbs. The extra weight is from the massive components such as the pinion shaft, crank pin assembly, and connecting rods, but the rotational mass is close to the Evolution.

The quality improvement appears to be on a level with S&S and JIMS flywheel assemblies. The structural rigidity of the Twin Cam flywheel assemblies is obviously stronger than any predecessor H-D flywheels. In fact, they approach or may even surpass the strength and wear limits of JIMS and S&S Evolution flywheel assemblies. This last statement is a mouthful to spit out as those of you familiar with these venerable aftermarket leaders probably regard their components as being the best in the industry.

The very reliable sprocket shaft tapered Timken opposed bearings arrangement has not changed since first use in 1956 right up until circa 2003 when the pinion roller bearing replaces them.

As mentioned, the left-side flywheel has slots machined into its outer rim in order for a magnetic sensor to read the

crank speed and position for both carbureted and fuel injected engines. This information feeds to the ECM where the fuel and linked spark tables determine fuel delivery and ignition spark. Pulse width or how long the injector's turn on for determines fuel delivery richness or leanness conditions relative to the air present.

The tables also known as the matrix, map or spark matrix map, is a series of cells with gradated spark and fuel settings determined by crank position, throttle position, engine load from manifold air pressure (MAP), and other sensors allow for more precise fuel delivery and ignition timing.

Earlier electronic ignition modules are like an on-off switch with zero adjustability; spark timing either retards or advances.

# Flywheel Problems

Harley-Davidson repeatedly teaches their technicians to replace the whole flywheel assembly if necessary.

Early on, some needed replacement because of the internal floating thrust washer(s) chewing between the flywheel halves and the connecting rod/bearing cages around the crank pin. Mostly, others have needed replacement because of the pinion half not truing correctly with the sprocket flywheel side with a resultant, destructive out-of- true pinion shaft. Flywheels rarely need replacement because of bottom-end bearing and crankpin malfunctions. Out-of-true pinion shafts wreak havoc on softer more vulnerable components that it encounters as it extends though the pinion support bearing, interacting with a fragile oil pump and cam support plate pinion bushing. Something will give, which is always the weakest part(s) and one of them will be the oil pump with its close-tolerance fitment.

The pinion shaft end that protrudes through the pinion bushing in the support plate will also not play out well, enlarging the relatively softer bushing. This fitment of shaft to bushing requires a close tolerance as oil feeds through this bushing into the pinion shaft. The oil needs pressure to reach its goal of lubricating the crank bearings. A sloppy and enlarged fit will reduce oil pressure to the heart of the engine.

In our experience at Heavy Duty Cycles, out-of-true flywheel pinion shafts are usually in the .003”- 004” range. Harley-Davidson is revising its acceptable truing specifications upwards. The above tolerance was not acceptable with the first Twin Cams or any predecessor engines. In 2008, this out-of-true specification of .003” to .004” is now okay! In fact, flywheel replacement does not occur now until reaching .005” tolerance.

Will New Replacement Cranks be True?

This may be okay with the Factory but if I may be so presumptuous to say, it is not okay with me on two counts; first, I want any pinion shafts I encounter to reduce to a maximum of .001” tolerance. Secondly, I do not want to subject my customers to a total flywheel assembly replacement cost when re-truing the existing wheels will accomplish close fitment specifications. How do I know, the new flywheel assembly is within the tolerance of .001” that I want for my customers? Perhaps I will be replacing one out-of-true flywheel set with a brand new one with the same problem!

My previous experience is that the vast majority of crank assemblies present no problems at .001” runout or less. In my

opinion, the tolerances are growing slacker because rubber mounting the Baggers and Dynas and counterbalancing the Softails mask the increasing vibration generated by these slacker tolerance specifications.

Another cause of flywheel replacement evolves from the replacement of the tried- and-true opposed-tapered Timken sprocket shaft end bearing assemblies circa 2003 with the huge roller bearing, first in successful use to support the pinion shaft side of the crank. I use the term circa because although the parts manuals delineate 2002 as the last year usage of the OEM 9028 Timken bearings, I find them in some 2003 models. Transition years often have a mixture-use of obsolete and new parts as old inventory depletes. This roller bearing (OEM 24604-00), presents potential endplay issues of the sideways movements of the flywheels in the engine casings.

The first symptoms of this endplay issue occur with

damaged compensator and alternator parts attaching to the end of the sprocket shaft.

Endplay is serious stuff as the crank can wander back and forth inside the engine casings eventually blowing up the engine if the churning flywheels make contact with either inner casing.

Now, let me put Twin Cam flywheel assemblies into perspective. Other than the above-mentioned cases, the flywheels are the best and the most reliable ever in the history of Harley. However, this is small comfort when it is your flywheels acting out.

## Twin Cam Flywheel Support Bearing(s)

### Sprocket Shaft Bearings 1999 to Present

The 1999 to 2002 left side engine-case-bearing(s) support the left flywheel sprocket (power), shaft. These bearing sets (OEM 9028), are inward facing tapered Timken bearings, consisting of two races and two bearings with a crank end-spacing shim between them. The shim determines endplay of the flywheel assembly. Endplay defines as how much the flywheel assembly can float back and forth inside the engine casings. The sixteen different sized shims vary in size from .0895” to .1195” in successive gradients of twenty-thousandths of an inch each.

Successfully used in all Big Twins from 1956 to 2002 these long-lasting bearings kits are very durable. The 24604-00, B, C, and D bearings are a big roller bearing measuring 2 OD and an ID of 1 In my opinion, they are suitable for the up and down vertical pressures imposed both on the pinion and sprocket side shafts of the crank extending through the engine casings.

However, I feel they are ill suited for the lateral horizontal-pulsing power surges traveling sideways from the crank to the primary compensating engine sprocket. Please refer to the picture Comparing Twin Cam Flywheel Support Bearings in Chapter I: The Platform for Performance that shows both a history of sprocket shaft and pinion shaft bearing use in the Twin Cam engine.

# 1999-2009 Sprocket Shaft Bearings

## Twin Cam Pinion Bearing

The first OEM 24623-99 pinion right-side engine case support bearing for the right flywheel pinion-shaft is a humungous roller bearing first used in 1999 on the Touring and Dyna Twin Cam 88A models. The bearing has an outer diameter (OD), measuring approximately 2 ¾” with an inner diameter (ID), of about 1 The individual 12-rollers are big, .700”-long with a diameter of .400”.

## H-D Part Number Suffix Use

The suffix A is short form for Alpha and B for Beta. They are used to distinguish the externally rubber mounted “A” engines from the internally counterbalanced “B” models.

The 1999 to 2002 pinion bearings used in the Touring and Dyna Alpha engines, 24623-99, 24623-99A and 24623-99B represent three refinements to the same basic bearing. The part number without a letter attached to the end is the initial bearing use by Harley-Davidson. The first change to the 24963-99 bearing was necessary to combat a whistling noise caused by piston descending-ascending air transfer between bottom end and timing chest engine compartments. Engineers reduced the fitment of the roller in its cage from approximately .006” to .022” after Sept 25, 1998 to between .002” to .008” to eliminate this transfer noise. The refined bearing now identifies as OEM 24963-99A. A further refinement resulted in the bearing nomenclature to change from OEM 24963-99A to OEM 24963-99B. Thus, the three OEM 24623-99, OEM 24963-99A, and OEM 24963-99B

bearings all saw use in the Touring and Dyna Alpha engines from 1999 to 2002 until the introduction of the smaller 24604-00 series of bearings that also saw continuous refinement.

I have never seen one of these earlier larger bearings wear out in a properly cared for engine. The times I have replaced this Twin Cam pinion bearing was due to other parts self-destructing whose debris infected these rolling behemoths. Timing chest components are the usual suspects for infectious debris.

The Motor Company began using a smaller but still large pinion bearing, the OEM 24604-00 series with the introduction of the Softail Beta engines in the year 2000. Use of this smaller but still large roller bearing began in 2003 in both Touring and Dyna models.

This roller bearing has a 2 OD and an ID of 1 The inner diameter is the same as found in the OEM 24963-99 bearings because they both support the same size outer diameter pinion shaft. There are two more individual rollers in the OEM 24604-00 bearing than found in the OEM 24963-99 bearing. The individual 14-rollers are the same length as the 24623-99 bearings at .700”-long but with a narrower diameter of about .300” or about .100” less.

## 1999-2009 Twin Cam Pinion Bearings

## Twin Cam Flywheel Misalignment

### The Crank Pin

The three-hole oiling crank pin was a welcome feature that the aftermarket has been using forever. I document elsewhere the massive sizing of this crank pin relative to what went before.

The 1 ⅝” diameter crank pin has a feature that few knew about, which is a hollow center oil reservoir. It is a total departure from anything ever used on a Harley in its entire history. Technicians do not have the opportunity to take the crank apart as problematic flywheel assembly’s return to the Factory for warranty replacement reasons.

H-D inspects and analyzes warranty cranks for any possible impending problems that require addressing. Sometimes it is less expensive to give grudging warranty rather than redesign and manufacture a new component.

Even mechanics that work on Twin Cam engines all the time mostly have not had an opportunity to look inside the crank assembly. H-D has no listings or publicly available diagrams for separate crank parts. The reader can view a disassembled Twin Cam crank diagram in Volume I: The Twin Cam, Chapter I: March 1998: Introduction of the Twin Cam plus pictures interspersed in both Volume I and this Volume II.

Harley only sells the flywheels as a complete unit. When one starts to think it, through, are they even rebuildable? Once apart and pressed together again, the interference fit of the flywheels will not be the prescribed .007” press. There will not be a tight enough fitment as some mating metal is scraped away both in the removal and the reassembly process.

Rebuilding and the Crank Pin Interference Fit

A big problem for rebuilding these cranks is that the crank pin is hardened while its mating flywheels are not. They are relatively softer cast iron. If hardened, there would be less peel of material in the flywheel holes both during assembly and disassembly, which reduces the interference, fit specifications

There are three main ways of dealing with this lack of rebuild-specified interference fit other than resorting to a completely new crank.

Chemical Weld

One is the usage of adhesives; space age adhesives. Loctite and other chemical adhesives will more than make up for a few thousandths of an inch. If visualizing the words chemical the concept is easier to grasp. The mechanic must use heat

and cold to re-press the flywheels together on the crank pin ends. Even a .005” press interference fit will peel metal away during the press process if mating component parts are at room temperature.

Using Temperature to Maintain Interference Fit

Contracting the crank pin in diameter sizing by freezing cold, while heating the pin apertures in the flywheel halves is necessary. The two flywheels’ crank pin holes will require heat expansion to increase sizing. This will get the initial assembly interference fitment below .005” to prevent metal peeling and retain the integrity of the parts after they return to equally even and normal ambient temperatures.

Application of the chemical adhesive is before reassembly. The adhesive sets as temperatures normalize.

Another way of maintaining correct interference fit would be with oversize crank pin ends but not the center part, where bearing and connecting rod big end fitment is concerned. The captured caged bearings will fit over a crank pin end that is a few thousandths of an inch bigger as will the connecting rod big ends. However, if placing the caged bearings inside the con rod big end journals beforehand, they will not fit over the enlarged crank pin end.

Therefore, the bearings should be installed first and then the con rod journals, which are not a big deal, just an order of assembly.

This rebuild technique brings with it all sorts of technical problems. First, it requires very skilled mechanics. Those from

yesteryear like the Panhead era when parts fitment was an art because of inexact manufacturing specifications would be suited to this work. But then again, so will modern PHD technicians if H-D institutes new training and practice sessions at Harley University.

In the old days, crank pins and flywheels came differently sized, the result of imprecise manufacturing as different operator machinists made most everything on lathes.

Imprecise machining combined with the vagaries of inexact heat-treating relative to final sizing technology of the day made for interesting and sometimes challenging assembly techniques.

The disassembled Twin Cam flywheel's crank pin holes may not be perfectly round. In this case, the holes will need to be re-bored to a round fitment size with precise tooling

available in any machine shop or sophisticated mechanical service department. It cannot go too big or the flywheel half will need replacement. For the technically proficient, this procedure would be simple.

Heat and cold fitment is again necessary to prevent metal peeling during the pressing process.

Welding the Crank Pin to the Flywheel Faces

The third way is to weld the crank pin ends in four equidistant spots with stainless steel rod and Tig welding. This is common in drag racing circles and works very well.

The procedure chosen will depend on circumstance.

There is no point in splitting the flywheel halves unless necessary. If only needing re-truing after checking bearing fitment and con rod side play, I do not recommend disassembly of Twin Cam flywheels.

This statement does not apply to any predecessor engines, as I would recommend the opposite.

With the Twin Cam, I do not want to disturb the press fit any more than necessary especially if there is less than one hundred thousand miles of usage. All internal crank parts are bigger and stronger with the con rod big ends and the crank pin surrounded with more and larger bearings. The materials are superb right down to the captured bearing cages that are copper-coated for easier break in.

These bottom ends are good for a couple of hundred-thousand-miles. Perhaps we will even run into a few that

accomplish half-a-million miles sometime in the future.

## Hardened Flywheels Assist Press Interference Fit

As I have said, the H-D flywheels are relatively soft to the hardened crank pin.

This allows flywheel, crankpin holes to enlarge because of metal peel during disassembly and assembly, thus reducing crucial interference fit.

Let us look at another recreational sports industry where interference fit, straight crankpins, and mating flywheel halves are rebuildable. Snowmobiles have hardened cranks with the straight interference fit hardened crank pins. These hardened flywheels press apart easier and with less metal peeling than a hard crank pin fitted to a relatively softer

flywheel. If a snowmobile has a .002” interference fit, when pressed apart maybe only .0005” of the interference fit compromises bringing the fit down to .0015” when repressed back together.

These pins are then Tig welded to their flywheel half at opposite points of three and 9-o’clock or six and 12-o’clock. Some prefer to Tig weld the pin to the flywheel without further tadoo. Others like to die-grind a small trough at the juncture of crank pin to flywheel; weld in the trough and grind off any high points. If clearancing between the flywheel and engine casing is an issue, this is definitely the way to go.

These machines have less bottom end torque than does Harley-Davidson and are able to assemble with a lesser interference fit. However, they are easier to rebuild because of the flywheel hardening.

S&S Cycles, Inc. has special crank pins that weld on one side and press-interference-fit on the other. For now, their use is for race bottom ends. Racing usually tests future stock engine mass production parts. The weld in side has a shoulder that presses against the flywheel half for welding the two together. The other side is a normal interference press fit.

I would never think of these ideas for an Evo engine and certainly not a Shovel because of their rebuild intervals unless purely for a race application.

I know of the exceptions to what I am about to say, of sometimes mythical and sometimes verifiable high-mileage reports from some of these older Harley models. However, as a rule of thumb, a Shovel probably needs rebuilding certainly

two times and maybe three times per one Evo engine rebuild. An Evo, the most reliable Harley-Davidson up until 1998, probably needs a complete rebuild including bottom end crank at least twice for every Twin Cam total rebuild.

Riders staked their bragging rights on their Evo getting one hundred thousand miles before a rebuild. I think some Twin Cams will accomplish the five-hundred-thousand-mile marker before bottom end disassembly or replacement. That is not to say with any of the above examples that partial rebuilds, for example the timing chest or re-ringing the cylinders will be necessary along the way. The point is that the Twin Cam bottom end is so superior to what has gone before.

Welding is an option with high mileage where disassembly is a rare occurrence. In addition, if the welded side is on the left-power-side-flywheel and the press side is on the less-torquey pinion side then we have a potential winner here for engine longevity and much less risk of flywheel scissoring.

Then we combine this one-side weld with hardened flywheels.

I predict, with aftermarket component parts that it will not be long before rebuilding Twin Cam cranks will be normal for the progressive mechanics and shops.

The only way to combat this is if complete cranks lower substantially in pricing.

Of course, many people such as me will not tolerate sloppy runout specifications no matter the pricing advantage.

Flywheel Truing Specifications

Truing the flywheels is the next issue requiring fast thinking, fast hands, and lots of experience.

Harley specifications call for runout at the flywheel rims to be between 0.000” to 0.010” at the rims. I can go for zero runout, which is a mechanic’s dream but a full ten-thousandths of an inch (.010”) is way out of line in my humble, yet experienced, opinion. At Heavy Duty Cycles, we attempt to true our flywheels to a maximum of only one-thousandths of an inch (.001”), at the shafts. We try to true the rims to .001” also but this is sometimes unrealistic although it is easier now than in yesteryear because of excellent and precise manufacturing techniques.

In other word, the flywheel halves are identical in circumferential sizing and preciseness. However, I will not allow more than .003” rim runout, which is the time honored maximum dimension going back many decades. It is unacceptable for us to not to attain .001” tolerance or less at both engine shafts. At Heavy Duty, we never rebuild a set of flywheels with so much runout as now is acceptable to

Harley-Davidson.

Vibration is the immediate result, with premature wear of associated parts coming not long after.

It is true that any vibration is not as noticeable with either a rubbermount system or a counterbalancer system dampening it. In these cases, one component is creating the problem while the other is masking it.

In my experience both JIMS and S&S, flywheel assemblies rarely have more than .001” at either location. Many times, it is less than .001”.

Flywheel rims may be closer to each other on one side than the other may. They may also be out of true and cocked forward and back relative to each other. Checking runout is

necessary at both at flywheel rims and at or as close as possible to the ends of the sprocket and pinion shafts. Check flywheel rim runout on the left rim on the outer side because of the interfering cutout timing indents on top.

I thought I was reading a misprint in the workshop manual for the Factory's specification limits, so I checked three different years. I did not misread the manual at all. The Factory says to true flywheels to between zero-thousandths of an inch (.000"), and two-thousandths of an inch (.002"), runout at the flywheel shaft adjacent to the flywheel itself. 10

It is easier and sometimes more practical to measure here because the bearing surfaces are concentric. The same manuals instruct replacing the flywheel assembly once runout reaches .003" although the 2005 Touring Models manual extends this to an incredible .005". The 2006 Dyna Models

service manual brings this flywheel assembly replacement specification back to .003”.

Methinks the Motor Company is experimenting back and forth and I for one do not like it.

The Model 2008 Harley-Davidson Service Manual Supplement Part number 99500-08 increases specification latitude once again. It instructs to true flywheel shafts to between zero-thousandths of an inch (.000”), and four-thousandths of an inch (.004”), and to replace the flywheel assembly at .005” runout! 11

The Revamped and Retroactive H-D Flywheel Trueness Specifications

Traditionally, the determination of acceptable flywheel specifications is done with the crank out of the engine and in

the H-D truing jig.

In 2007, the Factory first began a new way of determining excess runout while the engine remains together, which I will explain further on in this section.

H-D has revamped all traditional flywheel runout specifications upwards. Furthermore, the Factory made these specifications retroactive for all Twin Cams back to 1999 cautioning readers to make corrections to their service manuals. The specifications are the same as for the 2008 CVO 110 allowing up to .004” (.102mm), runout on the shafts on the bearing surfaces near their respective flywheels. Four-thousandths of an inch here can extrapolate to more at the end of the shaft.

The sprocket shaft end, inside the primary casing has a

great deal of weight attached, rotating at high speed. The alternator rotor, compensating sprocket and big nut will create much vibration with .004” runout on the shaft way back at the bearing surface by the flywheel. I feel that the runout will be much more at the extremity. This will also cause premature primary chain wear as it tightens and loosens with each rotation of the shaft, causing once again more vibration.

I am always discussing what happens with this type of exaggerated runout by the time it extends to the pinion shaft end, reaming out the pinion bushing in the cam-support-plate that depends on close tolerance fitment to transfer oil-feed to the bottom end bearings, crank pin and con rod big end parent material races. The poor victimized oil pump, half way along the pinion shaft. If the complete pump moved with the pinion shaft’s runout, wear would not be so much of a concern. However, the relatively soft aluminum oil-pump body firmly bolts to the stationary cam support plate on the

inboard side just before the pinion bushing. The hardened inner feed and return geroters attach to the rotating pinion shaft, which also drives them. The outer feed and return geroters float in their confining space, between the inner geroters and the soft alloy pump body. The outer geroters will force against the oil pump inner body by the errant runout on the pinion shaft carving away at the soft underbelly.

Obviously, H-D engineers are attempting to stay within runout bounds that will not destroy the oil pump and ream out the pinion bushing within predetermined obsolescent miles. However, it does not take a rocket scientist to figure out that the parts longevity of the oil pump and pinion bushing and hence the cam support plate compromises.

The rhetorical question becomes how much longer these parts will last if there is less unruly runout with which to

contend. I would venture an educated guess that parts longevity could easily reduce by half. Severity increases with increasing runout and diminishes with less.

Harley-Davidson has determined that the geroters will shatter inside the oil pump at around .015” (.381 mm), pinion shaft runout. 12 This is the Armageddon specification. Trust me; there is lots of wear and debris occurring long before reaching this specification.

The obvious symptom will be total loss of oil pressure. If oblivious to this symptom, the rider will begin to hear all kinds of noise as broken parts interfere with other moving parts and loss of lubricity will result in grinding and other stresses.

The Factory advises flywheel replacement at .005” (.127mm), runout on either of the bearing surfaces. Of course,

I think all of these specifications are far too much. It also looks to me like the Motor Company is experimenting with tolerances both acceptable to the Harley rider and for parts longevity.

It is too expensive to replace flywheels that are untrue when first installed in new motorcycles.

The bottom line is that the aluminum oil-pump housing and the brass cam-support-plate bushing are soft and cannot withstand an errant pinion shaft not trued as perfectly as possible.

Measuring Runout with the Engine Together

This procedure is new as of the July 26 2007 and circular Tech Tip TT234, Twin Cam Crankshaft Inspection & Run-Out

Specification describes it.

This procedure involves measuring runout on the sprocket shaft while the engine is together. This method accomplishes easily with the engine in the frame. Removal of the outer primary and the compensator package is necessary. The alternator rotor now also serves as the diaphragm spring package for the compensator engine-pulse shock absorber.

In all previous Big Twin engines going back to 1970, the alternator rotor is completely separate from the compensator package. The 2006 Dyna and all 2007 and later Twin Cams have a much different spline system on the end of the sprocket shaft. There are 24-fine splines as opposed to the ten-⅛”-wide splines on all Big Twin engines from 1956 to 2006 except for the 2006 Dyna series.

It is also easier if removing the inner primary but I have

found this unnecessary if being attentive to the runout measurement process. The rear wheel should be off the ground. Remove the spark plugs and put the bike in a high gear to make engine rotation easier by turning the rear wheel with your foot while observing the measurement process.

Only spark plug removal is necessary if the engine is on the workbench. At Heavy Duty Cycles, we take a female splined compensator part and weld it to a two-foot bar to make an engine-rotating tool. Slip the female spline end over the male splined sprocket shaft and it is easy to turn the engine smoothly.

Set up a dial-indicator attachment tool so it is rigid. A magnetic base assists setup of the dial-indicator attachment tool. Each setup is different, so it is up to the technician-mechanic to jury-rig it.

The Factory wants the dial indicator at and perpendicular to the sprocket shaft. I put it, anywhere convenient up to and including the 90-degrees since the goal is not to have the indicator probe that rides on the splines to dig in.

Furthermore, we are looking for the measurement runout from the top of the splines, ignoring or dismissing any generated by the valleys. My indicator probe end is of greater size than the fine splines so this is not a consideration for me. This procedure is more difficult but still doable on 1999-2006 Twin Cams with the much wider one-eighth-inch splines. Set the indicator to zero for ease of reading the micrometer gauge face. Any runout will likely cause the gauge needle to move to either side of the zero position. If it only moves one way, reset the gauge location and re-measure. It may be that the gauge probe placement inadvertently found the lowest spot so that

the gauge only moves to one side when rotating the engine. We want to ensure that the gauge is properly setup so we redo the placement just to be on the safe side. Add both maximum movement figures together (zero to the left and zero to the right), to determine total runout.

This requires concentration but is not difficult.

For example if the maximum needle travels .003” to the left and .004” to the right, the total sprocket shaft runout will be .007”. This runout will always be greater than if measuring sprocket shaft runout with the crank removed from the engine on the concentric shaft-bearing surface near the flywheel. This is why the following runout figures seem extreme relative to the rest I have examined with the flywheel assembly removed from the engine.

Zero runout at the sprocket shaft means perfectly trued

flywheels. Good luck but I do not think any mechanic will find zero runout, as this is too rough a measurement technique. However, the Motor Company deems runout up to and including .010” (.254mm), as acceptable. Any runout of .010” or less requires no further action.

For me, again the higher figures are undesirable.

I would advise to immediately remove the timing cover and pinion drive gear to measure pinion shaft runout before making a determination on removing the flywheel assembly from the engine casings for replacement or re-truing.

The Factory stresses again, not to re-true the flywheels because the integrity of the press fit of the crank pin to the flywheel halves will likely compromise. 13 I agree with the compromised interference fit. However, I will re-true the

wheels anyway if there is no damage to bearing surfaces or other crucial component parts. I Tig spot-weld the crank pin ends to the flywheels in four equidistant spots with stainless steel rod to ensure no further shifting out-of-true.

If the sprocket shaft runout equals or exceeds .012” (.305mm), runout, this will mean that the flywheel assembly will probably need replacement. A runout reading of this magnitude requires the disassembly of the engine for further testing of the crank on a truing jig to get an accurate view of runout.

If runout equals or exceeds .005” (.127mm), at the bearing shafts with the crank removed from the engine, then the crank needs replacement according to the Factory specifications. Again, I make a determination at that point if I feel I can re-true and save the assembly and expense of replacement. Moreover, how do I know the trueness of a new flywheel assembly set since there is much latitude in what is acceptable

to Harley-Davidson?

A follow-up circular TT352 dated December 3, 2007, Twin Cam Crankshaft Assembly Run-Out Inspection follows up the first one and describes how to measure but not correct the different out-of-true conditions with the crankshaft disassembled from the engine.

The six conditions are; flywheels scissoring one in front of the other, flywheels pinched together opposite the crank pin, flywheels spread apart opposite the crankpin, flywheels angled in tandem to the right, flywheels angled in tandem to the left and lastly, one shaft bent creating runout on one side only.

Disassembling Flywheels

Taking Twin Cam flywheels apart is no easy chore relative to 1998 and earlier. The .007 interference press fit of the crank pin ends into their respective flywheels is a super-tight fit that does not want to give way. Dislodging every other H-D predecessor flywheel from its crank pin is child's play compared to the way Twin Cam flywheels glue together. There is no locating half-moon Woodruff key, normally found between the crank pin and the pinion, right side, shaft flywheel half.

Up to 1999, the separate Woodruff key normally fits into a half-moon indent in the crank pin's tapered pinion side end in engines. When fitted into the flywheel, the flat sides of the key mate with a flat indent in the flywheel half. This ensures that the oil feed holes align on the pinion side and will help resist spinning the shaft in the flywheel on the sprocket shaft power side from sudden torque increases. Before 1999, pinion shaft keying was considered necessary to guarantee that the oil feed conduit that originates from the oil pump and feeds to

the inside the pinion shaft, mates with another feed channel in the pinion flywheel that feeds up to the crank pin. It is crucial that the holes all line up for oil delivery to the bottom end bearings, con rod races, and crank pin.

In early Evo years, both shafts became integral with their respective flywheels so the pinion shaft no longer needs keying to the right flywheel. Wisely, this innovation carries over into the Twin Cam. CNC machining of the flywheel and shaft casting ensures the pinion shaft is precisely at 90-degrees to the flywheel for crank trueness.

This is a great advance for the elimination of vibration.

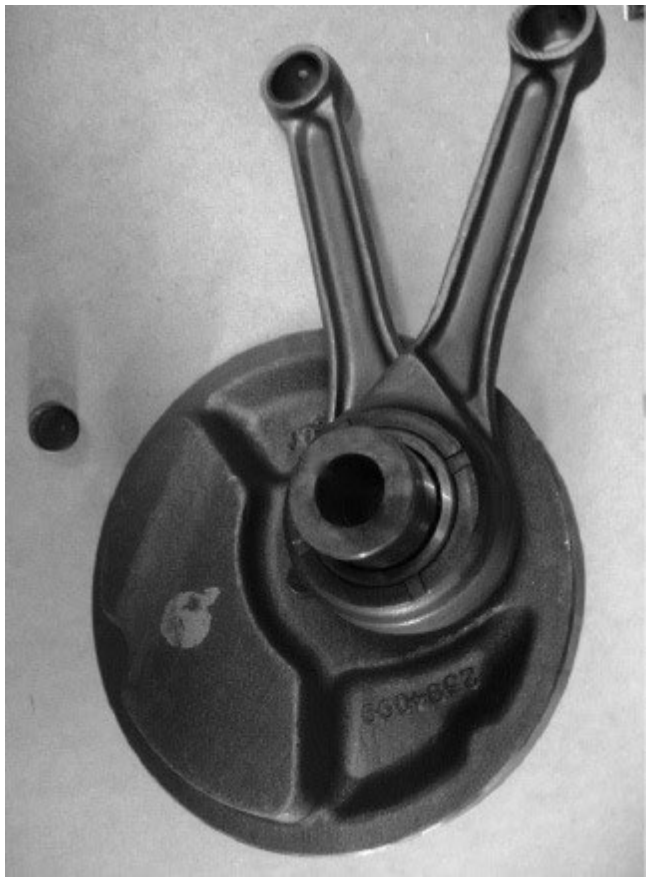
The abbreviation CNC stands for computer numerical control, and refers specifically to a computer controller that reads computer software instructions to drive a powered

machining tool that selectively removes excess material. In other words, the bulky rough casting machines down to the precisely made flywheel half. I cannot complement Harley-Davidson enough for this simple innovation made possible by progressive thinking and modern CNC machinery. CNC machining of integral flywheel components increases parts longevity and the lessening of vibration forces.

The same process applies to the previously separated sprocket shaft and mating left-side sprocket flywheel, which is the power-side flywheel.

The Twin Cam flywheel assembly is different for a seemingly similar component from anything that has gone before with Harley cranks.

Twin Cam Pinion Right Side Flywheel Half



Shown are 1999-2006 TC 88 pinion-side flywheel-half, with

crank pin and con rod bearings with male and female connecting rods with sprocket shaft flywheel removed. Slots machined into the sides of the con rod big ends where they encircle the outer bearing cages and crank pin to allow lubricating oil easy access in between the parts. The film strength characteristic of oil will always separate moving parts with a layer of oil sometimes only molecules thick. The lower male connecting rod fits between the two-pronged female connecting rod. Shown is the male small end rod piston-wrist-pin-bushing. It is big and muscular unlike its 2007 small-end counterpart, which narrows and points in order to reduce reciprocating weight. Reciprocating weight refers to parts that go up and down as opposed to rotating parts that turn. Pictured to the left of the flywheel is the crank pin end cap that press-fits into the end of the crank pin to create a reservoir for cooling oil. Oil carries away heat as it circulates back to the oil tank. This is one reason an oil cooler is a good idea for the Twin Cam engine so that heat may dissipate away into surrounding air. The raised “pork chop”

portion of the flywheel half in combination with the indentations above repositions the weight of the flywheel to create more torque so that bottom-end inertia is only about 70% of the Evo flywheels. This means there is less lag time when the throttle cranks for engine off-the-line response. This simple weight redistribution allows the Twin Cam to accelerate faster off the line than an Evo. The paint blotch on the pork chop is factory identification mark whose different colors may indicate such things as balance factors. The casting number in the flywheel identifies the part first manufactured in 1999, indicated by the last two digits, in this case 99.

The crank pin presses into each flywheel and attaches them together as one unit. The shafts are located in the center of each flywheel while the always-offset crank pin is towards the flywheel rims to create the This offset crank pin determines how far up the cylinders the rotating flywheels component connecting rods will thrust the pistons and then

pull them back down. If the crank pin places in the center of the flywheels in alignment with the sprocket and pinion shafts, the flywheel assembly will still easily rotate in its support-end bearings. However, the connecting rods, while still rotating on the crank pin bearings will remain stationary in the cylinder bores. The attached pistons will also remain stationary, not push up, and pull down their respective bores.

When the crank pin reservoir is full, pressure from oil still being delivered into the crank pin will force oil out of the three oiling holes to lubricate each set of three bearing cages with eighteen roller needle bearings each. The purpose of this largely unknown reservoir has an ingenious simplicity, which is cooling.

I initially took the trouble, actually it was fascinating, to disassemble a Twin Cam crankshaft. Now I do it all the time.

On previous engines, disassembly is easy. Zap off the crank pin nut on the sprocket shaft side and gently tap the flywheel half on the side below as far away as possible from the crank pin with a mallet made from hard rubber, soft metal lead, or brass. We do not want to damage the pin or its seating taper in the flywheel.

The flywheel separates easily from the tapered crank pin.

Well, it is not so easy with a Twin Cam. The biggest diameter crank pin to date press fits into each flywheel half at, a .007” interference fit. This fit does not seem overly tight, does it? It is extremely tight. The flywheel halves are not easy to separate. It is quite a job compared with earlier cranks to disassemble. In fact, a hydraulic press is necessary to separate the flywheels by pressing the crank pin perfectly straight and perpendicular to the flywheel face. This new disassembly procedure is a long way from a tap with a soft mallet.

There is a new type of captured bearing cage that prevents the bearings from rolling sideways and thereby creating drag as opposed to rolling true and straight. The cages are copper-coated for break in ease. There are three more bearings for a complement of fifty-four, (three sets of 18), compared with 51, (three sets of 17), in the Evo and late Shovelheads. Bearing diameter is the same as the Evo but these Twin Cam rollers are longer providing for more stability and longevity of service.

There is still the traditional configuration of the single-fork, front, male connecting rod and the dual-fork, rear, female con rod. The connecting rod big ends surround the roller bearing captured cages and rotate on them. Twin Cam big ends are massive when compared with predecessor big ends. Parent material substitutes for traditional pressed in big

end races. Parent material uses the connecting rod metal as a race, machining and hardening these three bearing surfaces into smooth precise races. Connecting rod big ends are blue from hardening. This is normal whereas in previous connecting rods would indicate extreme heat wear requiring replacement.

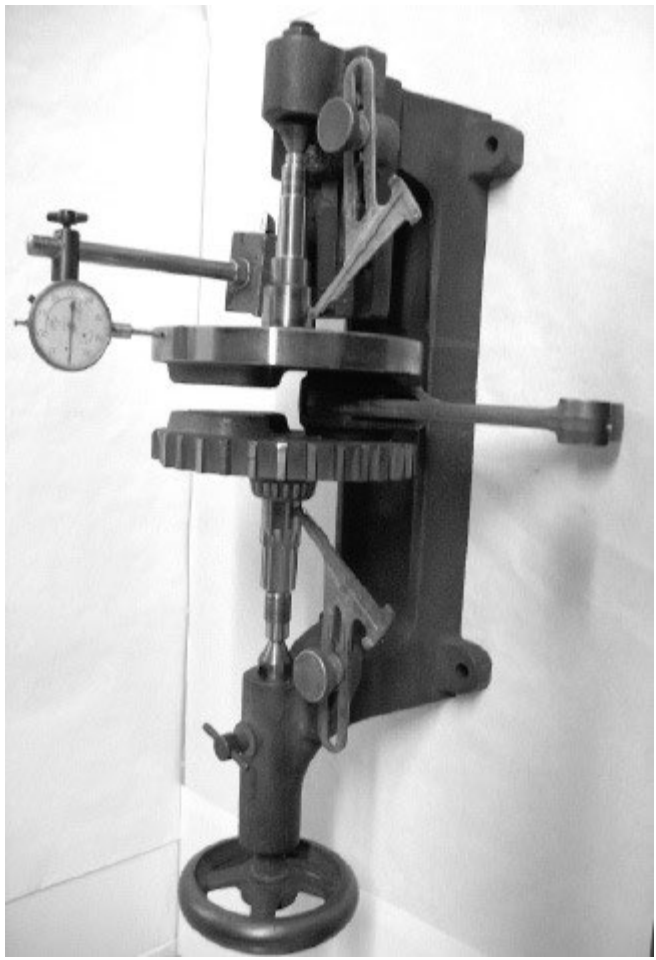
I have documented the massive piston wrist pins and small end wrist pin bushings as well as the bulky square sizing of 1999-2006 small ends relative to the tapered small ends utilized in 2007 and later Twin Cams in Volume I: The Twin Cam.

As I have said, Twin Cam bottom-end inertia is only about two-thirds of the Evo due to repositioning of the weight in the flywheels. There are pictures in both the first and the second volume that show this pork chop repositioned-weight inner flywheel casting. What this means for the rider is less lag time when cranking the throttle. Response is faster than with the

Evo.

The rigidity produced by these superlative parts produces less noise and vibration and is an excellent base for performance upgrades.

Twin Cam Crank on H-D Truing Stand



Harley-Davidson modified its truing stand (part number HD96650-80), in 1980 because the motor shafts became longer a decade earlier, making the flywheel assembly wider. The truing stand in the picture, part number 96650-30, works well for me even though it is the earlier version by 50 years. The 1999-2002 Twin Cam crank on the flywheel-truing jig is identifiable by a tapered Timken bearing and ten ⅛”-wide splines on the sprocket shaft plus the connecting rods are pre-2007 because the wrist pin small ends are boxlike in shape as opposed to tapered on the TC96 models. H-D advises not to use the centers on the flywheel shafts in the truing stand because they may not center I do not understand this because the centers must be accurate for flywheel machining after the casting process. Actually, I do understand Harley-Davidson’s concern since it is possible for the subsequent heat-treating to possibly distort the center-machining hole marginally. In any case, every set we have trued on the stand centers correctly although we must check every set because of the Factory’s advisement. The flywheels must turn freely in

the truing stand jig. The runout indicators will not read accurately if the flywheels sit too loosely or too tight in the jig centers. Of course, Twin Cam flywheels are not rebuildable because no parts are available. Furthermore, H-D does not want Twin Cam flywheels re-trued because the integrity of the crank pin interference fit may compromise. I spot-weld the pin to the flywheel face in four equidistant areas to ensure no movement of parts relative to each other after truing. I think that it is onerous for the rider to have to pay for a complete flywheel set, when out of true, instead of having a relatively inexpensive repair option. In addition, how does the purchaser know that the new flywheel assembly will not have runout up to .004” on the shafts with new sloppier specifications introduced in 2007? I want my flywheels in my bike and customers bikes to have .001” or less tolerance runout on the shafts to prevent destructive vibration induced forces. The truing stand runout indicators are spring-loaded and rest on a plate that has markings in two-thousandths of an inch (.1mm), gradients. The markings are far enough apart to

distinguish runout of .001” or less. The indicator is screw-locked down after placement against the bearing surface of each shaft close to the flywheel half. As the flywheels rotate on the jig centers, any runout will register on the indicator plates. These indicators will also adjust to fit against the flywheel rims to measure runout there also. However, the crank-position sensor cutouts on the rim of the sprocket shaft side prevent this for that flywheel. I will place a dial indicator against the rim flywheel side face, which will indicate some out of true like pinching and spreading runout but not up-and-down. I have a dial indicator micrometer set up on the pinion side flywheel to measure its runout. The magnetized base and adjustable arm allow easy access to measure in most cases. The heavy pork chop portions of the inner flywheels are visible. This weight repositioning will increase bottom-end throttle response.

Truing Flywheels

The reader will not find any information on truing flywheels in any modern factory workshop manual after the year 2000. The only 1999 manual to have this information is the Softail one because the engine is not Twin Cam but the single remaining Evolution model. There are no pictures or diagrams except what I present in Volume I: The Twin Cam and this current volume for the reader to view what component parts look like.

Please forgive me as I once again digress into some nostalgic history. In another of Donny's Blasts from the Past, I will look at mechanics over four decades ago.

Donny's Blasts from the Past: Fats, the H-D Mechanic

Back in the day, Fats was the man. He could true flywheels with his eyes shut. Fats never let a set of flywheels go until

the shafts were at the minimum runout possible.

This was in an era when sometimes the mating flywheel halves were not even the same size. Rebuilding Flathead, Knuckle, Panhead, and the very new Shovelhead flywheel sets was an everyday occurrence.

Fats and the service manager, Fast Eddie would eyeball the different size halves and align the smaller one in the middle of the other while rotating the flywheel assemblies in the H-D truing stand (part number HD-96650-30), intently watching the mechanical measuring arms move back and forth indicating the type of runout needing correction. Fats and Eddie judiciously whacked the flywheel halves with big brass or lead hammers so as not to dent the rims into a best compromise relationship with each other. Truing flywheels was an art as much as a science back then.

Riders of the day rebuilt their bottom ends sometimes on a yearly basis. It was part of the ritual of riding as mechanics and working on your bike was every bit as important as actually riding them. Your average rider could disassemble an engine. Adjusting point-condenser mechanical ignitions and cleaning lead (gasoline), and oil-fouled spark plugs became second nature, as did adjusting chains and those dammed mousetrap clutches.

The new luxurious foot shift models gradually began to replace the suicide foot clutches.

It took awhile longer before the external spring-return throttle cable superseded the internal handlebar dead man's throttle that stayed in position to exactly where it was turned. Many years later adding the second idle cable prevented the lone throttle cable sticking wide open. Now we have

electronic fly-by-wire throttles eliminating cables altogether.

A clutch cable connected the clutch lever on the handlebar to the big booster spring, part of the mousetrap, half way down the front left frame down tube. From there an adjustable steel rod connected to the clutch arm on top of the transmission. The clutch arm extends over to the right side of the bike and down into the kick-starter cover to push the clutch rod. The clutch rod extends internally through the transmission main shaft back to the left side of the bike to push the clutch springs closed, disengaging the clutch plates or letting them rebound to engage by forcing the plates together with each other. An improperly adjusted booster spring could spring at any time usually when your fingers were in there adjusting away. The mousetrap severely pinched fingers or rapped knuckles.

Speaking of clutches, riders became expert at disassembling them on the side of the road, scraping the oil off them by standing on each plate with one foot and rotating them on the cement or asphalt to remove impregnating oil. Dry clutch plates will not tolerate oil as they will slip and not grab.

Today a clutch plate will not work for long unless bathed in oil for cooling. Wet clutches were not an option back then, as the oil would leak out all over the place.

After the ritual engine disassembly, riders of the day marched into Mouldy's to have Fats or Eddie rebuild and true their flywheels.

Fats would be smoking reefer out back in the service area, lazily doing a perfect job on a set of flywheels. Mouldy, the dealership owner would start screaming about what stunk.

“Cigarettes Chief”, Fats would drawl, with content half-smile as he went about his work.

Everyone called the boss Mouldy behind his back and Chief to his face. Mouldy was cantankerous but a true motorcycle enthusiast as was everyone back then. When is the last time you saw the owner of your dealership riding? Like him or hate him, Mouldy could wheel a rigid frame through its paces with its gas tank hand-banger or transmission jockey shift, suicide foot clutch and dead man’s throttle…but then so could everyone else.

Truing Twin Cam Flywheels

Steve, myself, and our machinist Marvelous have begun truing errant Twin Cam flywheel assemblies with vigor with our combined 100 years experience in H-D mechanics.

Steve and I are using traditional truing methodology such as fixing the shaft ends into the centers in the H-D jig while Marvelous uses the centers in one of his lathes to fit in the ends of the Twin Cam pinion and sprocket shafts. The Factory warns against this. They say the shaft indents may not center to the crank pin axis, which would be impossible anyway since it is hollow thus giving erratic readings with the flywheel jig truing indicators.

Harley-Davidson advises against using its traditional truing jig, in use for over a half-century because of this. I do not understand this because the shaft end manufacturing indents need centering for machining after the casting process.

I will check every single crank we do for off-center indents but so far so good, on both early 1999-2006 and 2006 plus all 2007 and later flywheel assemblies. Therefore, the mechanic

must pay attention for this incorrect centering if using a traditional truing jig with Twin Cam cranks.

If finding this off-center condition, follow the Harley-Davidson measurement recommendation.

H-D Rationale for not Trusting Shaft Center Machining

The reason H-D does not advise using the flywheel shaft end-centers for runout determination is possible distortion during heat-treating. However, they center perfectly for machining the flywheel and shaft together before the hardening process.

During hardening, the shaft centers do not move but the center-hole may distort slightly. This is not a big deal until this distortion multiplies along the length of the shaft.

It is conceivable but does not always happen that this distortion may add up to a few-thousandths of an inch to the runout figure.

In fact, in our experience at Heavy Duty Cycles, it has not been an issue but it could well happen. However, when the Factory is saying that .012” runout is acceptable what is a few potential thou. I say all this because H-D is technically correct when advising mounting the shafts onto bearing rolling centers and not the traditional end-centers of the H-D flywheel-truing jig.

On the other hand, this raises another issue that will affect runout determination potentially giving false readings once more. If the journal centers support by ball bearings, the play between balls and cages will allow distorted runout readings.

The same problem applies to roller bearings for the same reason. In fact, this applies to any type of bearing. The correct journal-rotating center must be very precise oilite bushings with extremely tight rotational tolerances to eliminate runout distortions from the test equipment.

I mention these points to make the reader and/or mechanic aware of these possible aberrations and to make allowances for them.

However, the fact remains that all pinion and sprocket shafts including all pre-Twin Cam crankpins have centering holes that may have distorted from the hardening process for the last one hundred years! This would mean that millions of crankshafts, in fact every crankshaft, may have been incorrectly trued for the whole of Harley-Davidson history.

Harley-Davidson currently advises suspending the crank

pinion and sprocket shaft bearing surfaces on rollers. This is possible in the bed of a lathe. Two dial indicators are now necessary, one to measure each shaft's runout. 14

In scissored flywheels, we are finding other complicating out-of-true condition(s) also. We re-align the scissoring condition first and then re-true the other condition, whether it is pinching, spreading, or flywheels leaning in tandem.

I will discuss this in detail a little further on.

I replace the flywheel assembly if finding a bent shaft since re-truing is impossible and actual breakage may be imminent. Of course, as time goes on I will acquire good flywheel halves from other damaged cranks for rebuilding purposes.

I have questions that revolve around how pinching,

spreading and leaning can occur with a straight-fit crankpin and massive machine-straight flywheel crank pin holes. As I have said, nothing much has changed in 105 years of truing or aligning the flywheels with each other in order to reduce vibration except up until the mid and late nineteen-eighties when pinion and sprocket shafts were cast integral with their respective flywheels.

This major simplifying change makes truing flywheels very much easier than previously. Before this integration of shaft to flywheel, the truing process began by aligning the shafts at exactly perpendicular-true to their mating flywheels. Many in the past inexplicably ignored this first and very important step.

How these two flywheels align to each other has everything to do with the amount of vibration that will result.

Furthermore, the outer shafts, one for each flywheel that extends through each crankcase also has much to do with flywheel alignment or trueness. Each shaft supports the flywheels as they rotate on bearings located in the crank case halves.

It bears repeating that in the mid nineteen-eighties, Harley began to cast first the left side sprocket shaft as part of the flywheel. A few years later circa 1989, the right side pinion shaft also became part of the cast flywheel. Again, this was a tremendous change in the history of vibration as the shafts could now be cast and then finish-machined to be perfectly straight at to their respective flywheels.

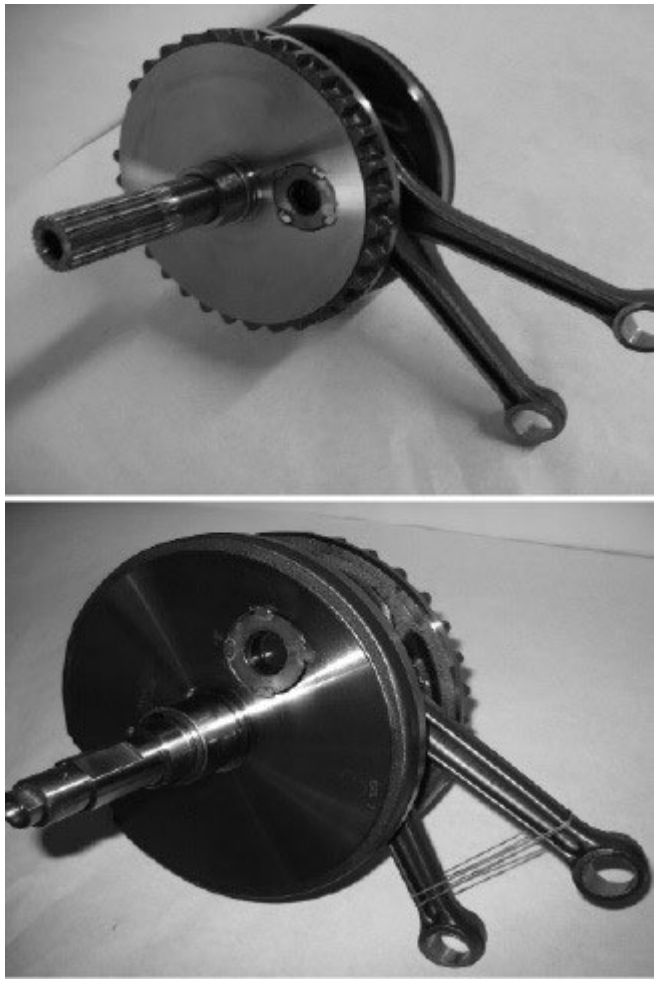
Prior to this, when installing separate shafts into the

flywheel halves the mechanic had to ensure that keyways used to center and add extra strength of shaft to flywheel fit properly. The problem of not removing burrs and resultant off centering of the shaft to flywheel can be a royal pain. This type of care separates an ordinary wrench from those that get the deserving reputations for fast reliable Harley's.

Burrs and/or improper fitment force the shaft to cock slightly so it is impossible for the shaft to fit at a perfect 90-degrees.

It is also time consuming, which leads some cost conscious consumers to the lower quote where none of this stuff is checked.

CVO 110 Flywheel Assembly with welded Crank Pin



This crank pictured top and bottom is out of a CVO 110 but it is the same as the TC96 crank and its big bore cousin the TC103. The flywheel assembly is different from the 1999-2006 cranks in three main areas. The first is the spline configuration on the sprocket shaft (top). Cast with and integral with the sprocket shaft is the left-side flywheel. The 1999-2006 apart from the 2006 Dyna, shafts have ten ⅛”-splines, which have been the same since 1956, compared to the 2006 Dynas and all 2007 and later Twin Cams where the splines are much thinner and increased to twenty-four splines. Therefore, engine sprockets, their compensator springs, and alternator rotors that sit on the sprocket shaft splines inside the primary case system are not interchangeable from 2007 to earlier engines or vice versa. The left side flywheel also known as the power side or the sprocket shaft flywheel has thirty equidistant ⅝”-wide indent notches about deep cut into

its outer rim. The peaks are about a ¼”-wide. Then there is one notch (7-o’clock position) cut the same depth but about 2 ⅛”-wide to complete the whole circumference of the flywheel. A sensor, which in this case is a magnet, sits in the front engine casing. The magnetic pickup of the sensor relays pulses generated by the notches rotating past it to the electronic control module, which will spark the ignition. This sensor can read where the pistons are in the cylinders and the position of the cam lobes as well as when the valves are opening and closing because of the sequential series of key parts throughout the engine that are timed to each other. The second way is to look at either the TC88 or the TC96 connecting rod small ends. Small ends refer to the smaller ends of connecting rods that house the piston wrist pin bushing. Big ends (not visible), refer to the much larger opposite ends of the connecting rods that house the crank pin and bottom end bearing rollers plus cages. TC88 connecting rod small ends in use from 1999 to 2006 shown in the previous picture titled Twin Cam Pinion Right Side Flywheel

Half are much larger than the new TC96 ones shown in these pictures. They are roughly square with the wrist pin bushing being the same length all around instead of tapering. Obviously, the TC88 con rod small ends are much stronger than the new tapered TC96 ones. Harley has made this change in order to reduce weight in order to reduce the reciprocating weight. This refers to parts that go up and down like the connecting rods and their attached pistons. Less weight means less stress especially when these parts must suddenly stop at the top (TDC), and the bottom (BDC), and then start up again going the opposite way. Lessening reciprocating weight was necessary to counter the extra stresses imposed by increasing the stroke since the shorter pistons must go up further in their bore and back down further again but must do it in the same time; therefore, the pistons and con rods must do it faster. The third way to distinguish early and late flywheel sets is the stroke determined by the position of the crank pin relative to the flywheel rims. The 4 ⅜" stroke TC96, TC103 and CVO 110 cranks locate the crank pin closer to the flywheel rims than

does the 4.000” stroke TC88 and TC95 flywheel assemblies. The stroke increase radius is whereas the diameter is the full ⅜” difference in stroke. The bottom picture shows the opposite right side of the crank or flywheel assembly. Cast, as one piece and integral are the pinion shaft and pinion flywheel. The TC88 pinion shaft is 40% thicker (.400”) than its Evo counterpart at the pinion-bearing location. The rotating pinion shaft ultimately drives the oil pump, cams, and the valve train. Going from the flywheel out, the thickest portion of the shaft is actually a pressed-on race further held in place by a circlip. The biggest pinion bearing located in the engine casing in the history of Harley supports this section of the shaft. The TC88B Softail pinion shaft has a sprocket mounted on it before this race to power the balancers and balancer chain. The next step-down section is outside the bottom-end compartment and is the first portion protruding into the timing compartment on the A engines. The S&S Reed valve installs here for problematic blowby engines. After the next step-down is the slotted portions on both sides of the

shaft, where similarly slotted oil pump geroters sit on the shaft. The rotating shaft drives the oil pump. The oil pump body housing the geroters mounts to the inboard side of the cam support plate. The next smaller diameter section with the oil feed hole sits in the pinion shaft bushing located in the cam support plate. The oil pump feeds oil into a similar feedhole in the bushing and into the pinion shaft for delivery to the crank pin, bottom-end bearings, and con rod races. The pinion shaft finally sticks through the cam support plate with a slotted end on one side only. The primary cam-drive sprocket, similarly slotted, mounts onto this section and a bolt that threads into the end of the pinion shaft fastens the sprocket in place. Both top and bottom pictures show the four equidistant stainless steel Tig spot welds on this re-trued flywheel assembly to ensure no scissoring will take place with future usage.

Flywheel Shaft Concentricity with its Bearing Surface

While this section is not applicable to the Twin Cam or late Evos for that matter, the information will round out the reader's lnowledge base. If rebuilding a Twin Cam crank, check the shaft to flywheel alignment. It is almost certain that alignment will be right on but it never hurts to check.

Something that yesteryear H-D manuals never advised but S&S Cycle, Inc. addresses is checking the concentricity of the shaft's taper with the bearing surface of the shaft. If these are out, the flywheels can never be trued accurately leading to a vibrating, bone shaking ride.

Individually install each shaft into a lathe so it can rotate with no side movement. Using a dial indicator check the concentricity (roundness) of the taper on the shaft. A dial indicator is micrometer that checks a moving rotating surface for roundness or concentricity.

Now, realign the indicator to measure the same concentricity on the bearing surface of the shaft. If the concentricity of either surface is out-of-whack relative to the other, by say one-thousandths of an inch (.001”), or more then use another shaft.

When truing the mating shaft to the individual flywheel at 90-degrees, this .001” difference will extrapolate into more runout with proceeding down the shaft to its end. The shaft will need replacing to gain better results. Furthermore, if either surface, in itself is out of round then replace with another shaft.

Just to be clear, there are two problems to look for; the first is concentricity of the taper relative to the bearing surface and the other is for either the bearing surface or the taper to be out of round. S&S shaft specifications dictate a tolerance of less than .0003” (three ten-thousandths of an

inch), runout on the ground taper, and the bearing surface by far the best specification of any separate flywheel shaft I have seen. All of the above might seem like minuscule stuff but we are talking about flywheel runout at the rims a full 5-inches away of more than three-thousandths of an inch creating a Milwaukee

Any small runout at the center shafts will extrapolate into much greater runout at the flywheel edge. The flywheels can never true accurately if these related surfaces are out of concentricity with each other,

The following truing procedure relates from Evolution models back into history.

I am taking this information from the 1940-1947 Knucklehead and 74/80 Flathead manual. Engine sizes are

1200cc in the case of the 74 and 1300cc, which translates into 80 cubic inches. This is not a mistake, as the 80 cubic inchers were also known as 1300's and not 1340's as is the case with the Evo 80 cubic inch. The Knuck came in 61 cubic inches (1000cc) and 74" (1200cc).

Here we go. "Wipe the pin taper and flywheel taper perfectly clean and free of oil. The manual is not kidding. Bearing fitment requires honing the con rod big end races and there is abrasive grit everywhere after this procedure. Install crank pin and key in the right side (pinion shaft) flywheel. Tighten (crank pin), nut very using Harley-Davidson special wrench." 15

Man, how do you like that statement? Very tight, makes me nervous!

The 1980-1983 FLT/FXR manual lists the torque for 1980

and early 1981 as between pinion nut torque 120-160 ft-lbs. and the 1981 and later nut at 140-170 ft-lbs. 16 You need more than a wrench to attain these torque ratings. Other conflicting manuals give higher torque ratings for the same year and shaft. After torquing the right side nut down, install a lock washer with a screw into a threaded hole in the flywheel to prevent the nut from turning. “If necessary, tighten nut a trifle more to make lock screw holes match” 17 Compare this stuff with the modern precise fitment I have been describing.

Next, check the oil feed passage through the pinion shaft, right flywheel, and crank pin with compressed air. “After correct connecting rod bearing fit has been attained, thoroughly clean all parts, and lubricate with engine oil (50 weight), preparatory to assembling flywheels. Install connecting rods on crank pin bearing so female (forked end) rod will be to the rear and male (single end) rod will be to the

front.” 18

You may hear old time mechanics talk about an engine running backwards. What they are referring to is an engine running the way it is supposed to except someone put the connecting rods in backwards, male to the rear and female rod to the front. The correct fitment is the opposite with the single-fork male rod always in the front cylinder and the female dual fork rod to service the rear cylinder.

After fitment, place the right side pinion shaft in a vice with copper jaw inserts to protect the shaft. Then install the left flywheel with both tapered parts thoroughly wiped of oil and grit onto the crank pin and lightly tighten the crank pin nut. While doing this; “align as nearly as possible concentric with right wheel by means of a straight edge held against outer face of wheel rims, from crank pin.” 19

Tighten the crank pin nut some more while checking alignment with the straight edge. As the nut gets tighter, it will be necessary to correct rim position with a lead or copper hammer. Do not use a steel hammer, as it will dent the rims.

At this point, it will be necessary to assist the flywheels to remain stationary in the vice as the mechanic tightens the crank pin nut. “Insert a rod approximately ½”-diameter and at least 5”-long through holes in flywheels and shift flywheels in vice jaws so that the rod bears against some part of the vice.” 20 After the “nut has been turned fairly tight” install the flywheels into a Harley-Davidson truing jig. 21

I learned through trial and error experience that the nut should torque to approximately 100 ft-lbs since the manual never told me. The early nuts are not as beefy as later year flywheel nuts that can be tighten further than 100 ft-lbs since

they were of stronger construction.

The truing jig is like a manual lathe where the jig adjustable pointed centers fit into and hold the slightly indented ends of the flywheel shafts. There are indicators, one for each flywheel or shaft as the case may be that are adjustable. The flywheels must rotate freely in the jig; too loose or too tight fitting and inaccurate readings will result.

The judicious use of the non-steel hammer will begin the alignment process. I do not strike the flywheel rims while in the truing jig. Rotating the flywheels will cause the indicators to move showing where they are still out of true; that is after you learn the ABC's of misalignment. The early manual does not describe this so I will jump to the 1980 manual, which does.

The “A” situation is where the rims opposite the pin are too close together while those near the pin are spread out. Hammer a hardwood wedge between the wheels where they are too close while the persuader again gently taps the rims near the crankpin. I can tell the reader right now that a hardwood wedge will not budge Twin Cam flywheels that pinch together opposite the crank pin. I use a homemade hard alloy wedge to get faster results and still protect the flywheels. The alloy is softer than the cast iron flywheels so as not to damage them via denting.

The “B” misalignment is opposite to “A” where the flywheel rims just above the crankpin are too close together and correspondingly the rim areas furthest from the pin spread too far apart. This corrects by placing a big C-clamp being on the rims opposite the crankpin while lightly tapping the rims closest to the pin with the persuader. This is awkward and not easy to do. I also do not like hitting the flywheels anywhere near the crank pin.

I substitute this H-D approach with my own.

Place the spread portions of the flywheel assembly between copper or aluminum protectors in a strong vice. Close the vice tighter. Experience will educate the mechanic on the amount of pressure to apply for spread amount and crank pin torque. Tap the vice with a steel hammer near the clamping area. The shock will help the flywheels align.

The “C” misalignment is where the flywheels skew or scissor from each other. During assembly, we try to correct scissoring with the straightedge. However, a few thousandths of an inch can easily escape this tool.

Progressively tightening the crank pin nuts during the truing process can also cause slight skewing. The lead mallet

comes in very handy here as I strike the highest side of the flywheel about 90-degrees from the crankpin.

Things really get fun, as the artiste must come out to fix the wheels when two of three of the ABC's are out-of-whack at the same time. A and B conditions cannot occur in the same set of flywheels because one is a pinching condition while the other is a spread condition. When this occurs, as it usually does, correct "A" or "B" before "C" while tapping the rim of the misaligned flywheel first. After aligning "A" or "B", proceed with correcting the "C" condition. 22 I true flywheels the opposite way, correcting "C" before "A" or "B". Then I check "C" again to ensure no shifting has occurred.

Some older flywheels take me many hours to get them as true as possible because of the non-precise flywheel sizing. JIMS or S&S wheels as well as the Twin Cam flywheels are so precise and smooth that sometimes trueness can take as little as fifteen minutes.

Both the Knuckle/Flathead manual and the 1980 FLT/FXR manual say that when striking the flywheels to loosen the truing jig centers a little. 23 Do not do too much or the centers may damage. Well, I guess there were some bad experiences with breaking pinion shafts over the years utilizing this loosening and striking procedure because the 1991 manual I looked in says, “Always remove the flywheels from the stand and strike the flywheel rim only at to the crankpin.” 24

Methinks the Factory keeps changing their mind on specifications and whether striking flywheels in the truing stand is a good idea or not. Any thinking mechanic would figure this out on his or her own. My opinion is not to hit the flywheels while in the truing stand nor hit them by the crank pin.

For a last piece of Harley instruction; “When truing the flywheels, the number of blows required and how hard they should be struck depends on how far shafts are out of true and how tight nuts are drawn…Never strike wheels a hard blow near the crank pin. This could result in a broken crank pin.” 25 Furthermore, I think it is a good idea to try to strike the flywheel rim in the area of 90-degrees from the crank pin whenever possible.

I have repeatedly chronicled the structural strength, quality and size increases of Twin Cam bottom end parts components. The rigidity produced by these superlative parts produces less noise and vibration and is an excellent base for performance upgrades.

The truing process is intimidating to traditional H-D mechanics especially in light of the fact that the Harley

manuals say to return worn flywheel assemblies to the factory.

Anyone with snowmobile engine expertise will adapt quickly. They already have experience with lesser interference fit shafts.

The truing procedure is much the same as with earlier Harley's utilizing crank pin nuts but it is of paramount importance that the .007" interference pin fit in the flywheels is maintained or substituted with chemical weld bonding agents or actual welding. The flywheels must press together on a hydraulic press, keeping care of alignment instead of drawn together by a tapered shaft and nut like on earlier models.

The aftermarket will develop crank pins for the Twin Cam

in progressively larger sizes to maintain the interference fit as the flywheel crank pin holes enlarge with successive rebuilds. The aftermarket Twin Cam stroker flywheels like those made by S&S and JIMS also use the interference, straight fit crank pin, and flywheel. I am sure JIMS and S&S will be at the forefront of rebuild parts for the Twin Cam flywheel assemblies.

Harley motorcycles have never vibrated less than they do today, but at the same time, vibration is part of the Harley Vibration is one of the unique ingredients that make up the whole of the Harley.

Let me discuss what is normal and what is not.

Some motorcycles have little or no vibration, but not our older Milwaukee For Harley riders, the goal is to control vibration, not eliminate it. Basic vibration is the by-product of

the engine design. As long as the engine design remains as it is, there will be vibration. The 45-degree cylinder offset in all Big Twins going back to early in the last century is the design factor that contributes to the vibration that cannot reduce.

This is one reason the V-Rod has a 60-degree cylinder configuration that eliminates much vibration. The V-Rod goes two steps further in that there are counter balancers inside the engine plus rubber mounts on the outside between engine and frame mounts thereby isolating the rider from vibration.

Changing Harley's Big Twin engine may be unthinkable, but there are things a rider can do to ensure buzzing keeps to a minimum. Balancing the flywheels will result a smoother running engine. Note that I said smoother, which makes for much more comfort but the some base vibration cannot eliminate.

At best, we are using band-aids to fix the symptom and not the cause. In addition, just because an engine has a reputation for shaking, it does not mean it is entirely at fault. Harley engines are guilty of some, but not all sources of putting your hands and butt to sleep. There are a number of other steps the rider can reduce vibration without ripping the engine apart.

Shimming or adjusting engine mounts to get proper frame/ engine alignment is one method. Dynamically, not statically balancing wheels and tires is another after ensuring they are in good condition. Other potential contributors are wheel sprocket/powertrain misalignment, worn handlebar rubber, and/or engine mounts, worn out shocks and not servicing front fork oil. Any one or a combination of those non-engine items can result in a Harley vibrating more than it should. If the running train (wheels, forks, etc.) and powertrain do not align, there will not be the smooth transfer of horsepower.

When one part works against another instead of with it, the result is vibration.

Even owning a relatively new bike is no guarantee against excessive vibration. Factory wheel bearing spacing can be off, so that is something else to check during normal wheel maintenance.

As a long-time shop owner, I can say that a minimum of 25-percent of all the mechanical parts sold is due to a lack of proper maintenance, installation, or care.

Sometimes it is difficult to tell what is normal. If not sure, ask an experienced rider or mechanic to take your bike for a ride. The motorcycle owner may be the poorest judge if anything is wrong with the bike. Problems can develop gradually over time. When this occurs, the rider will adapt to

the slow changes, not noticing a worsening condition as it creeps on. A fresh rider will spot something untoward immediately.

Existing vibration can also dampen by choice of foot pegs, handlebar rubbers, and handgrips. The fad of alloy billet foot pegs and grips are not conducive to damping vibrations. We have a standing joke at Heavy Duty that the less expensive foot pegs and grips are the most functional, long lasting and vibration resistant they will be. The stock H-D rubber pegs and grips are the best available and cost the least. The most expensive grips and pegs that can cost ten times as much put your hands and feet to sleep within 50 miles...but they sure look good.

I am not trying to ridicule anyone for following fads, as I am a victim of Harley fashion myself.

# Engine Balancing

Balancing the flywheels is an effective and a relatively inexpensive option when the engine is already apart for a rebuild.

In my opinion, the expense is prohibitive to take apart an otherwise good engine for the express and only purpose of balancing. I say this not to denigrate the process but to put balancing in perspective since most of us are on a budget.

Certainly, elimination of all vibration by balancing is impossible in a Harley-Davidson, as there are many causes, not the least of which is the awkward firing order of a V-Twin. However, balancing will substantially reduce vibration within a select rpm range. Balancing will also add much to engine longevity, particularly flywheel support bearings and

other structural crank supporting parts.

It is easy to deduce whether a vibration is from flywheel/ crank imbalance or from outside sources, I discussed previously. If the vibration occurs at the same rpm range(s) in every gear, then it is likely from inside the engine. If it varies, comes and goes, or occurs all the time throughout the whole rpm range it is usually an external problem.

Engine static or dynamic balancing is expensive unless the motor is already apart for a normal rebuild or hi-performance work. Static balancing requires a highly experienced mechanic. Dynamic or spin balancing requires the same mechanical skill in combination with a specialty balancing shop with expensive tools.

However, if replacing gaskets or doing some top end work

it is no big deal to weigh both pistons, (including rings, wristpins and clips), to ensure they weigh the same. Balancing has an awful lot to do with having equal weight going up and down in each cylinder.

Equal cylinder volume also assists greatly in less vibration. Older manuals have stated forever that putting an oversize piston in one cylinder is okay. Various manuals also state that an oversize H-D piston weighs the same as a smaller or stock one. In most cases, they are close but we at Heavy Duty Cycles have run across pistons of the same size that have dramatically differing weights.

Pistons weigh in metric grams. One thousand grams equals a kilogram, which converts to 2.2 lbs.

Equalizing piston weight will work, in any year Big Twin or XL but particularly in older engines where inequalities are

more common.

However, different size pistons result in less compression in the smaller cylinder since the cylinder bore has a smaller diameter. The swept volume in the larger cylinder is greater creating more compression with the piston at TDC (top dead centre). The combustion chambers remain a fixed size so the extra volume of air and gas swept into the larger cylinder bore by the ascending piston is greater creating more resistance or compression. This creates an imbalance in itself since one cylinder is working harder and producing slightly more power than the other produces.

Equalizing piston and related parts weight and volume size is easy for those that understand the dynamics but there is much more to balancing than just top end work.

It involves taking apart the crank, (flywheel assembly), and re-truing, (within thousandths of an inch on older models and tenths-of-thousandths of an inch on newer ones), both sides of the crank with the crank pin only while leaving out the connecting rods, bearings and cages. The next stage is to equalize the reciprocating weight, which I will discuss a little further on in this chapter. Usually, balancing the crank accomplishes by removing weight from the heavy left side or sometimes adding weight to the lighter right side of the crank.

After individually balancing the reciprocating and rotating motions of the flywheels, the mechanic reassembles the crank with the con rods, bearings, and cages. Most importantly, the mechanic must re-true the flywheel assembly again as close as possible to the trueness of the crank before balancing. This is painstaking work for sure, but your mechanic can explain what you do not understand, especially if there is a disassembled engine to observe during the explanation.

I do this with customers all the time. Mechanical functions are so much easier to explain while showing the parts and demonstrating their relationships with each other

Always check and adjust or replace all the potential external causes for vibration before going to the heart of the engine. It is a drag, to take an engine apart and dynamically balance it to cure a vibration…only to realize later, the cause was elsewhere such as a thrown front wheel balancing weight was the real cause of the vibration.

H-D Sprocket Shaft Runout Measuring Technique

In the background is part of the left side engine casing. The engine is still together. The wire coils form part of the alternator stator. The unevenness and sometimes spacing of the coils is normal. The alternator rotor with magnets lining its inner diameter rotates around the fixed stator that fastens to the outside engine casing. The 24-fine splines protruding from the engine on the pictured sprocket shaft holds the alternator rotor and the compensator package as well as the threaded on hold down nut. Fitting around the shaft is the sprocket-shaft-end-spacer sitting inside the engine seal. This runout measuring procedure is new as of the July 26 2007 and circular Tech Tip TT234; Twin Cam Crankshaft Inspection & Run-Out Specifications describes This procedure involves measuring runout on the sprocket shaft while the engine is together. This method is easily to accomplish with the engine in the frame. Removal of the outer primary and the compensator package is necessary. The pictured 2006 Dyna

and all 2007 and later Twin Cams have a much different spline system on the end of the sprocket shaft. There are 24 fine splines as opposed to the ten-⅛”-wide splines on all Big Twin engines from 1956 to 2006 except for the 2006 Dyna series. The rear wheel should be off the ground. Remove the spark plugs and put the bike in a high gear to make engine rotation easier by turning the rear wheel with your foot while observing the measurement process. Only spark plug removal is necessary if the engine is on the workbench. At Heavy Duty Cycles, we made an engine-rotating tool by using a female splined compensator part and welding it to a two-foot bar. Slip the female spline end over the male splined sprocket shaft and it is easy to turn the engine. Set up a dial-indicator-attachment tool so it is rigid. The magnetized base assists in setup. Each setup is different, so it is up to the technician-mechanic to jury-rig it. Harley wants the dial indicator at and perpendicular to the sprocket shaft. I put it, anywhere convenient up to and including the 90-degrees since the goal is not to have the indicator probe that rides on the splines to

dig in between the splines. Furthermore, we are looking for the measurement runout from the top of the splines, ignoring or dismissing any generated by the valleys. My indicator probe end is of greater size than the fine splines so this is not a consideration for me. Set the indicator to zero for ease of reading the micrometer gauge face. Any runout will likely cause the gauge needle to move to either side of the zero position. If it only moves one way, reset the gauge location and re-measure. It may be that the gauge probe placement inadvertently found the lowest spot so that the gauge only moves to one side when rotating the engine. I want to ensure that the gauge is properly setup so I will redo the placement just to be on the safe side. Add both maximum movement figures together (zero to the left and zero to the right), to determine total runout. This requires concentration but is not difficult. For example if the maximum needle travels .003” to the left to .097” and .004” to the right to .004”, the total sprocket shaft runout will be .007”. This external engine-sprocket-shaft-runout will always be greater than if measuring

sprocket shaft runout with the crank removed from the engine on the concentric shaft-bearing surface near the flywheel. This is the reason the following runout figures seem extreme relative to those found on shaft bearing surfaces when measuring the crank alone separate from the engine. Zero runout at the sprocket shaft means perfectly trued flywheels. Good luck but I do not think any mechanic will find zero runout, as this is too rough a measurement technique. However, the Motor Company deems runout up to and including .010" (.254mm), as acceptable. This requires no further action. For me, again these higher figures are undesirable. If finding a high runout figure on the external sprocket shaft splines I would advise to remove the timing cover and pinion drive gear to measure pinion shaft runout before making a determination on removing the flywheel assembly from the engine casings for replacement or re-truing. The Factory stresses not to re-true flywheels because the integrity of the press fit of the crank pin to the flywheel halves will likely compromise. I agree with H-D on the

compromising of the interference fit but I will re-true the wheels anyway if there is no damage to bearing surfaces or other crucial component parts. I Tig spot-weld the crank pin ends to the flywheels in four equidistant spots with stainless steel rod to ensure no further shifting out-of-true. If the sprocket shaft runout equals or exceeds .012” (.305mm), runout, this will mean that the flywheel assembly will probably need replacement. It also requires the disassembly of the engine for further testing of the crank on a truing jig to get an accurate view of runout. If runout equals or exceeds .005” (.127mm), at the bearing shafts with the crank removed from the engine, then the crank needs replacement according to the Factory specifications.

## Balancing: Reciprocating and Rotating Weight

Balancing is a specialized craft that is as much science as it is an art that effectively but not totally reduces vibration. Engine crank balancing involves the weight of two groupings

of parts. They are reciprocating and rotating parts.

Reciprocating weight refers to the weight of all the flywheel parts that go up and down the cylinders like the top portions of the connecting rods including the pistons, wrist pins, clips, and rings. In balancing, the reciprocating parts associated with and including the top portions of connecting rods we refer to, as the small ends must weigh the same. This is a logical process because if one reciprocating assembly weighs more than the other one, an imbalance of force will occur between the two while going up and down the cylinder. A reciprocating imbalance throws off a smooth functioning crank creating vibration and reduced parts longevity. Usually, the heavier con rod assembly has weight removed from one or more areas such as the thick wrist pin bosses incorporated in the piston, or the wrist pins themselves or if requiring minimal weight reduction, polished off the small ends of the affected connecting rod until both assemblies weighs the

same. Therefore, material is ground, machined or polished off each the piston's wrist pin boss, wrist pin end or the con rod small end to eliminate reciprocating motion imbalance.

The other component in engine crank dynamic balancing is the rotating weight. These parts do not go back and forth but spin or rotate like the flywheels, shafts, and crank pin and the connecting rod big Distribution of the mass on the inner rotating weight flywheel halves affects both balance and torque.

I will describe the pork chop weight addition to the inner Twin Cam flywheels that increases low rpm torque giving better throttle response than the Evolution flywheel weight distribution. When viewing a disassembled Twin Cam crank there is a raised portion of the inner face of each flywheel half in combination with indentations above that are much different from their Evolution predecessor counterpart. This descriptively named pork chop design repositions the weight

of the flywheel to create more torque so that bottom-end inertia is only about 70% of the Evo flywheels. Thus, there is less lag time when the throttle is twisted for engine off-the-line response. This simple weight redistribution allows the Twin Cam to accelerate faster off the line than an Evo

For diminishing vibration-balancing purposes, drilling holes near the rotating flywheel rims equalizes and balances weight distribution. Perhaps, this is where art takes over from science; as percentage balance factors are a hot debate topic amongst gear heads specializing in this field. I will explore this in more detail later on as I realize some preceding statements need expansion and clarification.

With the reciprocating weight component of balancing, the less mass that moves up and down the cylinder not only reduces internal horsepower losses but also relieves stress.

Imagine a piston barreling up the cylinder when suddenly in an instant it must stop and change direction, speed up only to repeat again a change in axis at the bottom of the cylinder. Piston racing, stopping, changing direction, racing, stopping, and changing direction occurs thousands of times a minute (rpm).

Amplified piston speed occurs with an increased stroke, since the piston must travel further but do so in the same time-period. The TC88 has a very safe stroke of four inches while the TC96 has a still safe stroke of 4 ⅜". However the longer stroke where the piston travels and extra ⅜" (.375"), up and down the cylinder on the ascending compression and exhaust strokes and the descending intake and power strokes amplifies the forces of the weighted reciprocating parts that must now travel .375" further but in the same amount of time.

The Factory wisely lowered the weight of some of these

parts. Of course, the flipside is that lowering reciprocating weight may make the parts weaker, reducing their longevity.

I cover this subject in detail in Volume I: The Twin which is a great source for background material.

TC96 Connecting Rods

To help reduce the longer-stroke, larger-displacement engine's vibration, the engineers reduced the TC96 flywheel assembly's reciprocating weight by designing new connecting rods and pistons. The weight reduction lightens in ways that according to the engineers that do not reduce their strength.

On viewing the connecting rod small ends that connect to the piston wrist pins and pistons, I have to take issue with the above statement although the redesign is still strong. The con

rod small ends taper from bottom to top on the 2007 models, taking away much material away when comparing to the squared full size 1999-2006 connecting rod small ends. They may be strong enough but in my opinion cannot be as strong as what they replace.

Importantly, any drop in the weight of reciprocating parts will allow for higher revving and less stress.

TC96 Piston Design

In combination with the connecting rods and their newly tapered and thus lighter upper small-ends, the piston design is like many stroker pistons. The wrist pin bores are closer to the oil ring lands, which combining with a shorter piston and ten percent less weight, for a 6-percent total reduction in the flywheel assembly's reciprocating weight.

The piston wrist pins are shorter than their TC88 counterparts and therefore lighter. As with any H-D piston, the wrist pins position in the middle of the piston. The wrist pins are always in line-parallel with the piston front and rear thrust faces.

The piston thrust faces are round to match the round cylinder that they move against separated by a film of oil. No modern H-D Big Twin and XL pistons that have gone before, contact the cylinder bore on the non-thrust faces. The areas between the TC96 thrust faces under the ring lands are flat. The 2007 pistons non-thrust surfaces move inboard much further in the quest for less reciprocating weight.

Earlier Evolution and Shovelhead pistons have a slight elliptical shape that the reader probably would not notice unless looking for it. The front and rear thrust faces are still

round but the sides have a slightly smaller diameter than the front and rear thrust surfaces. The Factory moves those inwards on the TC96 reducing piston weight since there is no contact with the cylinder bore anyway. The resultant lighter piston lessens reciprocating weight forces.

Hypereutectic TC96 4.375” cast stroker pistons are lighter than their stock TC88 4.000” stroke counterparts although forged pistons may equalize or increase weight. Piston length and therefore reciprocating weight usually diminishes in direct correlation with increasing engine stroke.

Wrist Pins

Inside these, two opposing TC96 piston side flats are the piston or wrist pin bosses, a built up area of support metal for the wrist pin to fit through. The wrist pin fits through one

piston boss, then goes through the wrist pin bushing located in the small end of the connecting rod and finally out the other piston boss. Circlips, fitting into grooves cut inside the outer piston pin bosses hold the wrist pin in place on both TC88 and TC96 engines. The pin must float in the wrist pin bushing, again separated by a film of oil.

The reason is that the pistons change direction slightly as they go up and down the 45-degree opposed cylinders. They need to be able to rotate slightly on the con rod bushing. The piston wrist pins and therefore the pistons themselves float in the wrist pin bushing.

The wrist pins fit into the piston bosses with a light hand press at room temperature.

Balancing

Truing and re-truing of the flywheels during the dynamic balancing procedure must not only be identical but to minimum specifications, not the current maximum allowed by Factory service manuals. I must ask the rhetorical question if sloppy and imprecise mechanics is permissible; what is the point of going to all of this precision work?

The flywheel assembly also known as the crank as well as the wheels on your motorcycle needs to rotate evenly about their central axis. However, it is not only the flywheels that need this balance but any rotating part or assembly in the Harley-Davidson from relatively innocuous cams, gears, sprockets and pulleys to parts that have greater mass and thus more potential for very noticeable imbalance. These include the front and rear wheels, the large engine-compensator-sprocket-assembly and the massive clutch pack.

Massive is an extension of mass. When any rotational moving part such as a flywheel assembly is out of balance, vibration will induce that causes damage to the component flywheel assembly itself and its immediate support mechanism, which in this case are the pinion and sprocket shaft bearings.

Damage is relative to the amount of imbalance and the strength of the individual parts. A state of balance defines as the moving mass evenly distributing about the rotating centerline so that any resultant vibration is at a minimum.

The balancing process is usually part of the manufacturing process but may also occur during a rebuild. The motivation for balancing H-D cranks is vibration reduction but the bigger picture is ensuring a longer service life for the engine.

As wear occurs so does the balancing requirements since

weight distribution changes. Balancing with liquid mercury, for example is self-adjusting to rotational speed as well as minute changes in weight due to wearing of individual parts. This type of balancer is an enclosed narrow ring with mercury inside that attaches to the circumference of a rotating part, be it a wheel, flywheel or a clutch pack. Mercury has properties that will adjust weight to the lightest area to even out the rotational mass as it changes rpm.

As H-D engines get progressively bigger through time from the Shovelhead 1200cc to the Evo 1340cc and the early Twin Cam 1450cc that graduates to the TC96 1584cc, the effects of imbalance become more severe. Of course, The TC96 is just the threshold to bigger engines as 124ci and 131ci are becoming common.

Balancing becomes increasingly more important. Balancing

needs for an engine change as rpm increases so that an acceptably balanced engine at low rpms will become unacceptable at high speeds.

Mechanics is all about compromise.

## The Formula for Centrifugal Imbalance

An imbalance condition produces centrifugal force, which increase as rpm climbs.

F = Force: R = Speed/rpm

F =

Therefore, the imbalance force increases by the square of the rpm. Simply, if the rpm doubles the imbalance force will

quadruple. Furthermore, if the rpm triples the force will increase by a factor of nine times. Lastly, not to belabor the subject but if the rpm quadruples then the force will increase by a factor of sixteen!

This increasing imbalance force causes the vibration that while annoying to the rider's hands, feet and rear end has a much more insidious result; vibration of the bearings and the supporting structure of the flywheels. Prolonged vibration results in parts longevity reduction and functionality.

Moreover, vibration transfers down the Powertrain affecting it negatively also.

Defining Dynamic Balance

I will further define dynamic balancing in two parts; first is

identifying inequities in the distribution of mass in a rotating object and secondly correcting the mass distribution in order to minimize the centrifugal force and the vibration it brings.

I cannot emphasize enough the damage, vibratory force imparts to supporting bearings and structural parts like say an inner primary casing. These casings have cracked and broken all through the stamped steel tin primary era before 1965. The aluminum primaries had breakage problems induced by flywheel, compensator, and clutch pack vibration from their inception in 1965 right up until the early nineteen-nineties when the Factory engineering finally made them resistant to these insidious forces.

Structural damage is certainly not contained to primary casings, as any rigid parts component will feel the brunt of vibration.

Twin Cam engine casings are superb and the strongest ever made by H-D vendors.

I need only go back to between 1989 and 1993 when the combination of poor engine case castings and vibration resulted in many a broken casing.

Geometric and Rotating Centerlines and the Principle Inertial Axis

The geometric centerline is the physical or actual centerline of the rotational mass (flywheels).

The rotating centerline of a rotational mass like a flywheel assembly defines as the axis the mass would rotate if not fixed by its supporting pinion and sprocket shaft bearings. Physics names the rotating centerline as the Principle Inertial Axis.

Inertia is a physical principle of mass. A mass will want to continue in its existing plane. If at rest, it will want to stay at rest.

Therefore, the principle inertial axis is the one the rotating mass would go to if not constrained by support bearings. When these two centerlines are not on the same axis, the rotational mass will be in a state of imbalance. If the rotating and geometric centerlines are on the same axis, then the flywheels, clutch package, wheels, or whatever will be in balance.

To take this concept a step further, imbalance is an uneven distribution of mass (weight), around its rotating centerline. There are different balance concepts and they define by the relationship of the geometric and rotational principle-inertial-axis centerlines.

## Static Imbalance

Static imbalance defines as when the principle inertial axis or rotational centerline is parallel to the geometric centerline. I will be describing how to balance flywheels statically in a subsequent section Statically Balancing Flywheel Assemblies of this chapter.

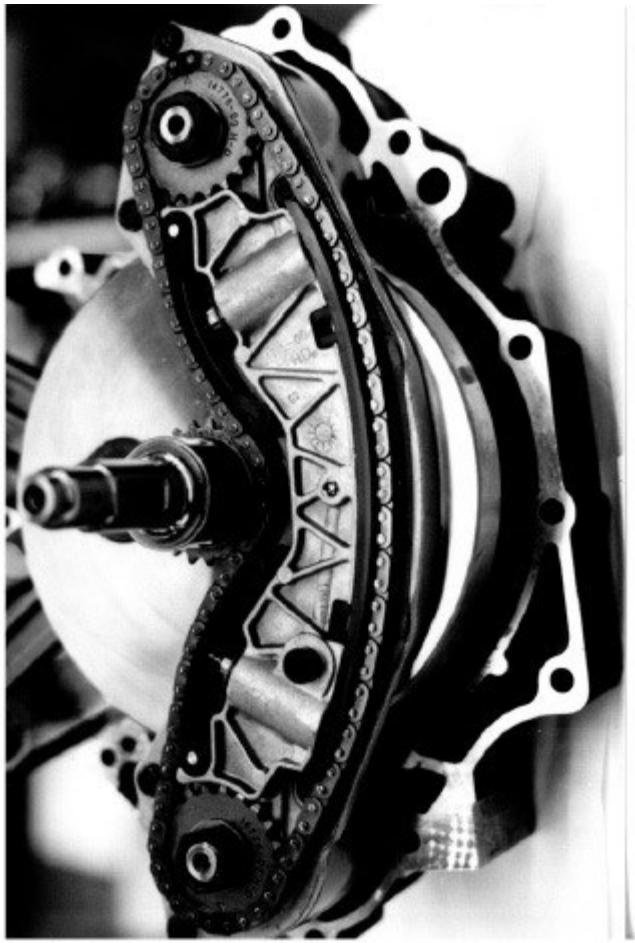
## Couple Imbalance

This imbalance defines as when the rotational centerline intersects the geometric centerline at the center of gravity.

## Dynamic Imbalance

As I have already said in a different way, dynamic imbalance defines as when the rotational and geometric centerlines do not correspond or intersect. Therefore, balance occurs when these two centerlines coincide with each other.

Twin Cam Softail Counterbalancer Assembly



Pictured is a counterbalancer assembly after removal of the right engine casing. Softail engine counterbalancers will wreak vibratory havoc if out of time with each other. This usually occurs with mechanic error. Bearing failure can precipitate an out-of-time situation but is an extreme case; always replace all four counterbalancer bearings whenever disassembling the engine. There are two bearings front and rear for a total of four with one in the crankcase (OEM 8989), and another in the housing (OEM 8991). The only other way for an out-of-time situation to occur is extreme wear. This may include the hydraulic adjusters collapsing; excessive sprocket and chain link wear or the chain missing a roller bushing(s). The chain would have to jump a sprocket tooth or teeth to bring the assembly out of time. Always replace chain and sprockets as a set during a bottom end rebuild. This is especially important with the counterbalancer mating parts, as there is much labor involved to get to them if requiring

service. Chains are demonstrative in their wear capabilities of mating parts. Sprocket teeth wear in the opposite direction of chain rotation. First, the teeth will narrow down from the top narrowest portion of the tooth. The narrowed top will then start to wear directionally as if pulled backwards. This hooking will increase until the tip breaks right off. The chain will begin cutting a new tooth, albeit a shorter one and the process will start all over again. This will continue until the chain will start skipping over the progressively diminutive teeth. The same will be happening to the other sprocket(s). The workshop manual says to replace sprockets if they show severe hooking. In my opinion, it is just too hard to get at them to allow any wear without replacement. At this point, the chain will also be badly in need of replacement, as it will be stretched very unevenly, rollers will be damaged, kinking may occur until a link finally breaks, snapping the chain. We do not want any of the above to begin happening in an enclosed engine casing as Armageddon may ensue. Always inspect the chain for missing roller bushings, side plates, and

adverse wear from turning, which may include burn marks. Once again, my credo is to replace, replace, replace when a wearing part is deep inside an engine or transmission. There is a sprocket at each counterbalancer end and the third is a center sprocket on the flywheel pinion shaft. There is a procedure for checking alignment of the flywheel pinion-shaft center sprocket to the balance-shaft. The balance shaft sprocket alignment tool (HD-44064) slides over the pinion shaft up against the center sprocket face sometimes called the timing chain gear. The alignment tool's appendage can now swing over the front and rear counterbalancer sprocket-face for alignment measurement purposes. Alignment must be within .014" (.356mm). Adjust the alignment if beyond specifications by replacing the spacer behind to offending sprocket with a correct dimension one. The Factory's addition of placement indicators, timing marks and color-coding achieves a simplifying methodology for timing the counterbalancers and chain. First, the following timing marks on the sprockets and timing link colors on the counterbalancer

chain must face outwards. Front and rear sprockets are stamped F and R for correct placement within the chain. With installation, always use new sprocket nuts and locks. With the F and R stampings at the 12-o'clock position within the chain, a timing mark will be at the 3-o'clock position on the front F sprocket, while a similar timing mark will be at the 9-o'clock position on the rear R sprocket. The center-sprocket-timing-mark should be at the 6-o'clock position on the pinion shaft. This involves rotating the flywheel assembly so that the sprocket presses on with its timing mark directly below the crankpin oil-feed hole in the pinion shaft. There are three colored chain links for timing purposes. The middle colored link in the chain must seat directly below the center-sprocket timing mark. The fore and aft colored links will now align with their mating sprocket timing marks. Counterbalancer shaft housings are marked F and R for correct placement in the front or rear engine-casing compartments. The balance shafts may be used front or rear but install with the smooth side down. Now insert the retention pins for the balance

shafts (HD-44062), inside the supports, fore and aft thereby locking the balance shafts in time with the engine. The center of the otherwise round pinion shaft flat for oil pump alignment is directly above the center-sprocket timing mark and below the crank oil-feed hole. It is essential that the retention/retaining pin in the end of the shaft does not become loose or dislodged. Consult the manual to perform a hydraulic tensioner leak-down test. There are two counterbalancer hydraulic tensioners, one before, and the other aft of the center sprocket. The tensioner comprises of three parts; a small cylindrical hardened steel piston, a tensioning spring, and a plastic vent cap. The hydraulic tensioner spring keeps the piston up holding the counterbalancer chain taut while at rest. Then check spring free length. It must be at least two 2.00” (50.8mm). The tensioner guides must not have any grooves or markings deeper than .090” (2.29mm).

# Statically Balancing Flywheel Assemblies

Dynamic balancing simulates moving parts in their running environment. The static balancing procedure occurs with the crank component parts at rest.

## Statically Balancing a Wheel

I will use static wheel balancing as an easier example to understand. I will then relate this to statically balancing an engine crank.

Static balancing a wheel is done by hand spinning the wheel to see where it will come to rest. If unbalanced, the wheel will stop in the same place every time with the heavy spot closest to the ground. Attaching a small weight…how

small is done by trial and error…to the opposite side of the rim to counterbalance the heavy spot will allow the rim to stop in a different spot each time it is spun. The wheel now balances statically.

Doing this to flywheel assemblies is more complicated but accomplishes with well-equipped shop mechanics tools. Remove flywheel weight from the heaviest spot as opposed to adding weight to the lightest spot.

Measuring and correcting reciprocating weight achieves in the same way as with dynamic balancing

The dynamic flywheel balancing procedure is not possible in an ordinary service department, as it requires highly specialized expensive equipment whereas a well-equipped service department with the expertise can easily do static balancing of wheels or flywheel assemblies.

I quote S&S Cycles, Inc. who sell a static balancing kit (51-1037), and have extensive instructions on its use in Instruction 51-1037; Special Shop Tools, Balance These instructions and a multitude of others are on their website (www.sscycle.com), under Technical. My quotations from S&S are not comprehensive instructions so please refer to their website for complete information. “The S&S balancing kit is designed to allow the engine builder to statically balance each flywheel separately in Harley-Davidson engines. The static method of balancing uses gravity to cause the heaviest part of the flywheel to rotate downward when placed on the balance stand. Weight is then removed from the heavy side by drilling holes. When the flywheel no longer rotates it is in balance….The balance factor we use is 60%. We arrived at this percentage after much experimentation, and found it to give the best results over the rpm range that riders most commonly use. Factors other than 60% will change the

direction and amplitude of the vibration. For example if the balance is at 50% or less, the engine will tend to vibrate vertically. As the percentage is increased, this vertical motion turns to horizontal motion. Horizontal motion from our experiments is more acceptable with less noticeable vibration transmitted to the rider. Therefore 60% is the percentage S&S uses as a compromise for best all round performance and riding comfort…The balance formula to calculate the weight of the bobweight that must bolt into each flywheel is:

$$\text{Bob weight per flywheel} = \frac{(60\% \times \text{reciprocating weight}) + (100\% \times \text{rotating weight})}{2}$$

Note that S&S uses 60% of the reciprocating weight. When we speak of the “balance factor”, we usually refer to the percentage of the total reciprocating weight used in balance calculations.” 26

The instructions are quite long but deal with the proper way to weight the small end of each connecting rod as well as the big end. S&S uses a 2:1 ratio in determining the weights of each end, the big end to small end because the big end is approximately twice as heavy as the small end. Then S&S explains the calculation methodology for bobweight weight. Next they show how to balance each flywheel and shaft individually. S&S discusses drilling techniques, safe areas to do so, depths and diameter holes to remove weight as well as the use of heavy metal (Mallory metal tungsten alloy), to add weight.

S&S demonstrates an invaluable calculation sheet and encourages mechanics to use it in every application. They also suggest keeping filled out copies over time for comparative purposes to discover similarities, aberrations, and trends in balancing work with H-D cranks. I reproduce this sheet below with examples of weights in grams and calculations for a 1987 Evo Big twin with stock bore + .005” oversize; 4 ⅝” stroker flywheels using S&S 7100 connecting rods and alloy bearing cages. 27

The procedure is identical for a Twin Cam.

However, leave some items blank such as nuts and nuts/locks/screws since there is none with integral shafts and a press-in crank pin.

Static Balance Calculation Chart

Please see the instruction sheets for further calculations and installation procedures for bobweight weights.

True Static Balance

Actually, my pal Jeff Davies showed me a way to accomplish static balance in a very precise manner with which, most mechanics are unfamiliar. First, the shaft centers must align true in a jig with the centers live, which means that the centers revolve on bearings so there is no resistance to movement. Perform this procedure without the reciprocating parts and the wheels true. Let the flywheels

settle to the heavy side in the jig, which they will do naturally. Find the center of the heavy side of the unbalance and extrapolate this center to the opposite side of the flywheel.

There are special tools such as a combination square and center-head-set that will assist the mechanic in finding the true center. A special holding fixture made from light material like aluminium or Delrin which is a plastic, will clamp to the center point of the light side of the flywheel exactly opposite the heavy center.

A fine, precise fish or pull scale now attaches to the fixture at ninety-degrees exactly.

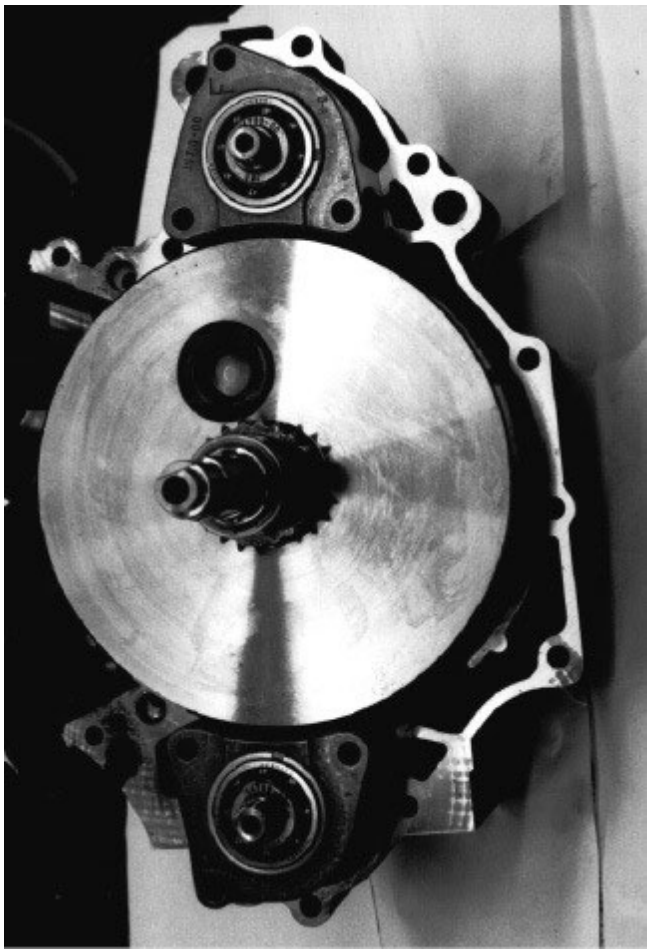
Pull the scale, always at 90-degrees as the flywheel rotates. The weight pull will indicate the weight of the heavy-side

imbalance. In other words, this weight needs to add to the light side in order to balance statically the flywheels perfectly.

In practice, balancers will drill out the heavy side by this indicated weight imbalance as close to the circumference of the flywheel as possible on the side. However, the balancer can also drill into the peripheral circumference face in the centerline but this must be at exactly 90-degrees or another imbalance may create. If more than one hole is necessary, the hole-pattern must balance also. In other words, pairs of holes must be equidistant from each other around the true centerlines.

The mechanic will weigh the tool and remove that weight from the equation.

Twin Cam Softail Flywheels with Counterbalancers



Pictured is a set of removed Twin Cam Softail Flywheels and counterbalancers without the right side engine casing, chain guide, and chain. Shown are the front and rear counterbalancers, marked front, and rear with their balance shafts and support bearings. The Softail engine cases elongate with knuckle protrusions to hold and support the front and rear balancers. It was necessary to manufacture a completely new rigid mount frame to hold this new engine style with its attaching transmission. In fact, the transmission becomes part of the frame as well as a rigid support for the engine. The rear center down tube does not extend down to the lower frame members as in previous 1984 to 1999 Softails without counterbalancers. The down tube bolts to the transmission, which bolts both to the rear engine mounts and to the lower frame tubes. The engine center-counterbalancer-chain-sprocket fits onto the pinion shaft close to the right side flywheel face. The outside end of the pinion shaft has a slot for a mating slotted drive sprocket to slide onto and bolt to after it passes through the pinion bushing in the cam-support-

plate. This sprocket powers the cam rotation and consequent valve train movements via the primary timing chain that attaches to the rear-cam-drive-sprocket on the outboard side of the support plate. Above the pinion shaft on the inboard side of the support plate the secondary timing chain connects from an inboard rear cam sprocket to the front cam sprocket thus driving and rotating the front cam. Between the center sprocket and the front balancer is the flush-end of the crank pin that press fits into and attaches both flywheels. Of course, the crank pin rotates with the flywheels. The crank pin could be in any equidistant position from the center sprocket and the pinion shaft. The indentation in the end of the hollow crankpin shows the end cap plug. One press fits into either end of the crank pin to create a cooling oil reservoir for bottom end bearings, parent material connecting rod races and the crank pin itself. The reservoir does double duty by feeding oil into each of the three sets of bottom end bearings, one each for both female forked con rod races and a third wider set for the single male connecting rod race.

# Dynamic Balancing

I am a big fan of dynamic balancing. However, it should never substitute for poor machining, substandard assembly of component parts and most importantly slack truing specifications. The flywheel assembly must true as perfectly as possible. To be blunt, new runout specifications for Twin Cam flywheel assemblies of up to .004” runout at the pinion and sprocket shaft bearing surfaces must tighten up substantially to at least .001” or hopefully less before employing dynamic balancing to further reduce vibration. The more true a set of flywheels are, the more likelihood that the rotating and geometric centerlines will coincide as long as weight is evenly distributed at the flywheel rims during the manufacturing process.

The definitive aim is to reduce the uneven mass

distribution to reduce centrifugal forces, which in turn reduces vibration in the support structure i.e. the engine and powertrain.

Balancing is straightforward once identifying and measuring the imbalance(s). Identifying the imbalance is finding the area(s) that requires either weight addition or reduction. Measuring or quantifying the imbalance is determining how much weight to add or reduce.

A balancing machine, which can be a simple rotational device or a complex automated one where weight reduction accomplishes by computer generated automated drills or milling machines with welding units to add weight when computer instructed. The rotational balancing machine measures the imbalance of centrifugal force of the flywheels, trued as true as possible with the crankpin only without the

reciprocating parts like the small ends of the connecting rods, pistons, rings, wrist pins and clips along with the crank bearings and cages.

The big end portions of the connecting rods are rotational weight. This weight adds to the rotational weight of the crank minus reciprocating parts. Temporary balance weights can determine where and how much imbalance exists. Then affixing permanent weights will put the flywheels in the closest compromise balance.

More likely, in the case of Harley flywheels, drill holes will remove the appropriate quantified weight near the flywheel rims in the generating areas of the most centrifugal force.

## The Benefits of Dynamic Balancing

In review, the major benefit of correcting imbalance is of course, the minimization of vibration.

However, noise lessening from mechanical vibration is an advantage that many do not realize. It meshes with the engine noise with which we are all familiar. Any contributing noise reduction is a duo-fold benefit as it assists in soothing the EPA.

Structural stress reduction of the supporting engine adds to longevity to both the motor and powertrain components as the power pulses travel the length of the motorcycle to the rear tire where it contacts the asphalt. Bearing life, in particular benefits from balancing.

Absorption of vibration becomes the unwanted job for structural components. Any readers that have the pleasure of owning a Shovelhead FL or FX are all too familiar with

cracking gas tank tabs, breaking coil mounts and broken inner primary case mounting tabs.

These problems result directly from the imbalance of centrifugal force at the rims of the rotating flywheels. Modern Twin Cam rubber mounting and counterbalancing mask these centrifugal imbalances but this does not negate the destructive vibrations from weakening their support structures. This translates into fewer miles before maintenance and repairs.

Balancing will also minimize rider fatigue and maximize riding comfort. Ride a thousand miles in one long day on an in-tune rubbermount with a 6-speed transmission. Try the same on a Shovel with a four-speed transmission. I said try because I cannot ride that far in one day on a Shovel but I have done so many times on newer vibration suppressed machines.

Balancing simply and importantly adds much to the quality and enjoyment of the ride.

Lastly, energy consumption reduces because interrelated component parts work together in harmony instead of contending with negative forces that create more work.

## Talking With the Balancing Experts

As usual, I sometimes need to consult those with more expertise than myself. I may have trued flywheels and stuff for over three decades and have much intimate knowledge but sometimes I need to ask those whom I consider the specialists to clarify subjects that virtually no one outside their realm of expertise understands.

This time I turn to my old buddies Scott Garland and Tom

Hong at D. Garland & Sons (www.dgarlandandsons.ca), for their sage advice and instruction.

## Balance Factors

Balancing a two cylinder will require a starting balance factor of 52%, which will increase depending on the year H-D crank. For example, an Evo crank will use a balance factor around 56% whereas the heavier Twin Cam crank may approach as much as a 60% balance factor.

Earlier, S&S Cycles, Inc., states that they use a 60% balance factor for static balancing.

This opinion difference is indicative of the debate that rages amongst the balancing experts as to the best balancing factor of reciprocating weight to the 100% rotating weight

factor.

Scott and Tom have been balancing H-D cranks for as long as I can remember over 35 years or more. S&S is celebrating their anniversary and yet even those with this much experience have their favorite balance factors that work well for them.

H-D engines because they are pushrod operated and thus have a limited rpm range require between 52% and 60%, compromise balance factors. Both experts find common ground at the 60% balance factor although Scott and Tom feel that some cranks require less for a Harley-Davidson Big Twin or Sportster

However, an overhead cam motorcycle like a V-Rod that has a longer rpm range and has a more comfortable cylinder angularity and firing order will require a much different

percentage balance factor. Furthermore, if the rotating crank had no reciprocating weight and was 100% rotating weight, the balance factor is 100%. There is no such Nirvana as a perfect balance for a Harley-Davidson.

We can never eliminate all the vibration from an H-D engine because of the reciprocating weight, the awkward firing order of the V-Twin and only two cylinders. We must choose a compromise weight factor for the rpm range most commonly used, which will be part of the power band.

Now, idle will not be part of this range so while idle vibration may reduce a little because of the balancing it will still feel much like before. However, with some 8-cylinder automotive engines, a more suitable balance is attainable where smooth operation results throughout the whole rpm range.

So, what in the heck is a balance factor?

Well, we will get a little more complicated before everything falls simply and surely into place. If we use a starting balance factor of 57% on a Twin Cam crank, this will mean that we are using 57% of the reciprocating up-and-down weight components including the reciprocating weight of the small ends of the connecting rods. Do not including the rotating weight of the con rod big ends as the percentage balance of reciprocating to rotating weight

On a 2-cylinder, we need to compromise with a percentage of reciprocating weight to the 100% rotating weight.

Splitting the Con Rod Reciprocating and Rotating Weight

As previously mentioned, the big ends of the connecting rods are rotating weight while the small ends are reciprocating weight. The male and female big ends house the crank pin, crank bearings, and cages. The small ends comprise the smaller portions of the connecting rods that house the wrist pin bushings, which connect the pistons to the small ends via wrist pins and clips. I will weigh the small ends and ensure they have equal weight by polishing material off the heavier small end.

This sounds all very nice does it not? However, we have two problems here, the first of which are the two connecting rods are all one piece with the big ends and small ends at opposite to each other and connected by a middle portion.

Scott and Tom come to the rescue once again.

Balancing shop’s utilize a special tool that merges a jig and

scale together to weigh the rotating and reciprocating portions of the connecting rods. Weigh the small-ends from their ends while the big ends separately weigh from their ends also. The jig-scale, in effect weighs the rods from end-to-end.

Suppose a connecting rod weighs 1000 grams. If the jig-scale weighs in the small end at 400 grams then the big ends will weigh 600 grams. I always weigh both as a counter test to ensure accuracy. Never weigh one end and simply subtract that mass from the total weight to derive the other end's weight. If making a mistake, it will carry over into the whole balancing equation creating a potential vibration monster.

The second problem is the female dual fork con-rod-big-end is about twice as large as the single fork male rod. How do I make these two weigh the same? I cannot. Furthermore, I need not. This is part of the rotating weight and therefore is at a 100% factor that needs no correction...that is until reciprocating weight adds its percentage into the equation.

The overall balance factor compromises the 100% rotating weight factor with the up-and-down components.

The S&S instructions in Instruction 51-1037, Special Shop Tools, Balance Kit gives very detailed information on this subject.

I will now continue with other vibration reduction techniques.

Near the end of 2000, S&S Cycle, Inc., the venerable masters of big-inch performance modifications and builders of performance engines for the V-Twin adjusted a long time policy regarding Static (at rest), and Dynamic (spin), Balancing. There are now three ways to balance H-D flywheels and assemblies: static, dynamic separating rotating

and reciprocating functions and dynamic balancing of the complete flywheel assembly as a whole.

I reprint the S&S circular below under Balance

Balance Techniques

“In mid-August 2000, S&S began spin-balancing most production flywheel assemblies with an American Hoffman dynamic balancer. With this technique, flywheels, crankpin, rods, and shafts are assembled; the weight of the piston assemblies and other components entered into a computer, and the completed flywheel assembly spun to a preset rpm on the computer operated dynamic balancer. The computer determines how much weight must then be removed from the outside of each flywheel rim to achieve the 60% balance factor normally used by S&S. No additional disassembly/

assembly of the flywheels is required.

With our previous method of balancing, the 60% factor was obtained by weighing the rods, crankpin, and piston assemblies, and then removing the required amount of material from the inside surface of each flywheel. Afterwards, the flywheels and components were assembled according to normal procedure. No additional disassembly/assembly of the flywheels was required.

Why Change?

Our primary reason for changing balance techniques was to meet increased production demands while maintaining the quality that S&S customers rightfully expect. As a point of interest, flywheels balanced using our previous static-balance method were assembled and then checked on the dynamic balancer. The difference between the two methods was less

than two grams, or approximately 0.1% (one-tenth of a percent).

While production flywheel assemblies (complete assemblies assembled by S&S only) will normally be spin balanced, S&S will offer static balancing as a no-cost option when shop time permits. Spin balancing is available only with complete flywheel assemblies assembled by S&S. Unassembled flywheels will continue to be static balanced unless ordered unbalanced.

It is possible to determine whether flywheels balanced by S&S were statically or dynamically balanced by removing the timing inspection hole in the crankcase, slowly rotating the engine, and observing for balance holes. Balance holes are 0.500 in. in diameter and will be visible on the outside rim of the flywheels that were dynamically balanced. Balance holes

are on the inside rims of the flywheels statically balanced by S&S, and not visible through the timing inspection hole.

## Warranty Issues

In the past, S&S has discouraged dynamic balancing. Our objection was that sources who balance flywheels, mainshafts and crankpin without the connecting rods, spin the assembly to determine how much material must be removed from the flywheels and then diasassemble the flywheels for balancing. The connecting rods are then installed and the flywheels reassembled.

It has been S&S's observation that, if not correctly performed, the additional disassembly and reassembly procedure required for this method can alter the tapers in the flywheels, mainshafts and crankpin. This can change critical tolerances, make subsequent assembly and truing procedures

difficult if not impossible, and lead to premature engine failure by preventing the precise mating of parts that is required for maximum strength and durability.” 28

Taper expansion is a real world concern but this is the way it has been done for a long time. However, there are crude mechanics out there that under-torque or over-torque while using steel hammers that can wreak havoc with flywheel assembly integrity.

Of course, Twin Cam flywheel shafts are integral with their mating flywheel so this taper issue is a non-issue. In addition, Twin Cam crankpins are straight as are their mating female holes in each flywheel. This eliminates problems also, but creates some new ones.

The taper enlargement concerns by S&S extrapolate into

the interference fit issues in the Twin Cam. A hack will enlarge and therefore reduce Twin Cam crank pin interference fit concerns way beyond what thinking, skilled and concerned mechanics will.

## External Aftermarket Vibration Dampeners

### Radial Shock Absorber Balancing

The Fisher vibration damper replaces the compensating sprocket spring package inside the primary cover and requires an hour or two to install. I am unable to find Fisher on the Internet (2008), so I cannot give a referral web address.

This damper sometimes spelled dampener is a radial shock absorber that not only controls vibration but also creates a few extra horsepower in two different ways. While it is not a direct drive unit, it transfers engine horsepower more

efficiently back through the powertrain than the stock Harley compensator.

Anytime an engine moves harmoniously instead of against itself with vibration-induced power pulses, it will produce more power. No energy or frictional loss of horsepower occurs.

Fifteen years ago, I looked for a vibrator to test the Fisher unit. You know, a Harley that shakes your feet right off the floorboards and puts your hands to sleep. In this case, it was 1993 Softail Nostalgia. First, I diagnosed the problem to ensure it was not severely misaligned or unbalanced flywheels. I found (surprise surprise) loose motor mounts. Therefore, I left them untouched and away we went.

The engine dampener reduces torsional flywheel vibration,

which in turn produces less aggravation with the loose engine mounts.

I then tightened the engine mounts. The result was a very noticeable reduction in vibration.

The customer felt a subjective 75-percent reduction while Fisher claims up to 60 percent. I felt the vibration reduction was less but substantial nonetheless.

Actually, the term balancer is not exactly right in this particular case because it dampens torsional vibration rather than balance engine flywheels. Put simply, motor parts with different masses and weights must move at the same speed if they attach to each other.

The result is dragging force against motor direction. Oscillation occurs, much like a tuning fork. The damper

absorbs the torsional motion via friction plates that convert the movement to heat, which harmlessly dissipates.

Two other benefits result.

One is a slight improvement in gas mileage since the fuel induction especially carburetion acts more efficiently when it is not having the bejeesus shaken out of it.

Another and more importantly, is an increase in engine bearing life. Think about it.

Less vibration, or more correctly, a dampened and controlled vibration, allows the bearings to turn smoothly. Therefore, the dampers like Fisher work to control normal vibration associated with uneven firing of the V-twin and the torsional movements of unequal weights, shapes, and masses

inside the engine forced to move at the same speed.

## Mercury Balancers

Dynamic balancing is an expensive procedure for engines since it requires extensive work but it does go to the heart of the problem. Torsional vibration dampers like Fisher are an engine external fix that for most engines works quite well and is less than going inside the mill. Another option is mercury balancers that are even less expensive.

They also have use on the clutch package, wheels and on flywheels. These types of balancers have been in use on big diesel engines for a long time. Mercury balancers comprise of a thin flat steel disc approximately the same size as the compensator or clutch package. A hollow ring at its circumference contains Mercury that has balancing properties with rotational motion. The flywheel power side has a groove

machined in its circumference where a hollow ring containing the mercury affixes. A wheel may have a flat disc or more practically, a light hollow ring affixed to the rim with the balancing Mercury inside.

The total clutch weighs in at between 15 and 20 pounds depending on the year and model, which is a lot of weight to be hanging on the end of the transmission shaft spinning at various rpms in the rear of the primary chain case. So why doesn't the Factory balance the clutch appendage, or for that matter, the compensator sprocket, which weighs in at five or more pounds depending on year and spins inside the front of the primary chain case on the end of the motor sprocket shaft?

I have not a clue but in my opinion, they should. It is somewhat ironic since Harley rubbermounts a whole

motorcycle to cure the symptom of vibration instead of the

Dynamically balancing the flywheels, compensator, and clutch package would go a long way to curing the shakes, not to mention having a positive effect on engine longevity and horsepower.

Spinning clutch vibration will rise and subside according to the bike's revs just as with the engine.

Vibration will normally occur at certain rpms and not as much at others, depending on the specific cause and the imbalance factor of the parts involved.

Single Fire versus Dual Fire Ignitions and Vibration

For many years, H-D felt that single fire ignitions were not necessary. However, the Factory used single fire point's

ignitions on their Knuckle and Panhead racing bikes. Harley-Davidson began using single fire ignitions again on fuel-injected models in the 1990's. All Twin Cams mercifully utilize single fire.

Single fire ignitions differ from the traditional dual fire because there is only a spark for fuel ignition near the top of the compression stroke to drive the piston down on the ensuing power stroke. This single spark during the four strokes that comprise one engine cycle reduces a small amount of engine vibration as opposed to the dual fire systems in use for so many years.

Dual Fires have a wasted spark also on the exhaust stroke, which ignites unburned gases causing the engine to work against it.

# Conclusion and Summary

Performance and vibration control go hand-in-hand. Performance will suffer if coexisting with a bad case of the shakes. Moreover, performancing will exacerbate any existing vibration. Vibration will damage performance components.

Furthermore, excessive vibration will affect motorcycle handling. It is impossible to eliminate engine vibration from Harley-Davidson Big Twins including the Twin Cams plus the Sportster family and Buell's, except possibly the 2008 1125R.

The primary internal cause of V-Twin vibration is the uneven spark-igniting firing positions of the cylinders. A four-stroke engine like Harley-Davidson requires of crankshaft rotation to complete all four cycles or strokes each lasting crank rotation. Therefore, the two cylinders fire awkwardly, one-cylinder fires at while the other fires later.

Harley-Davidson has now liberalized allowable runout specifications for current models. Furthermore, the Motor Company has made these tolerant specifications retroactive back to 1999 to encompass all Twin Cam engines. A new methodology for measuring potential severe runout figures primarily due to a new phenomenon, scissoring of the flywheels institutes in late 2007 for measuring sprocket shaft runout with the engine still in the motorcycle. We now measure runout on the splines of the sprocket shaft as an indicator of whether stripping the engine apart for flywheel replacement is necessary or not.

With counterbalancing the Softail models in 2000, all Twin Cams now have a way of isolating the rider from the discomfort of vibration. Rubber mounting and counterbalancing engines mask the annoying symptoms of

vibration, which in turn allow the liberalization of flywheel rim and shaft runout specifications.

However, vibration has much more impact than as an annoyance to rider comfort. Vibration is insidious in reducing the life of support parts, in particular the support pinion and sprocket shaft bearings. I believe the huge roller bearings are massive for this very reason, to combat the effects of vibration. Support structures like the engine casings will also suffer a reduction in longevity, as superbly made they are because of runout liberalization specifications.

Shaft runout both affects the large and heavy compensator as it spins at high rpm in the primary casing creating vibration pulses down the powertrain as well as the delicate oil pump and the pinion bushing in the cam support plate directing lifeblood oil supply to the bottom end bearings and

crank pin.

Oil pump longevity suffers from increasing runout specifications as does the reaming effects of the pinion shaft end enlarging the pinion bushing, lowering oil pump pressure of oil.

I cannot overemphasize my beliefs that increasing runout tolerances will have a negative effect on engine and to a lesser extent powertrain longevity. In the final analysis, I think that the Factory hides the liberalized symptoms of vibration through counterbalancing and rubber mounting to the detriment of longevity.

It would be so much better to retain strict runout specifications in combination with vibration suppressing balancers and rubber. Imagine a Harley powertrain with strict flywheel truing specifications like those of forty years ago;

with dynamic balancing; with compensator and clutch package balancing, whether dynamic or the more practical Mercury rotational balancers along with either counterbalancing or rubber mounting vibration suppression.

The only improving addition to the above dream Power Train would of course, be increasing cylinder angularity to or but that is heresy.

Rider comfort would increase once again.

Furthermore, the Twin Cam, the most reliable Big Twin Harley-Davidson to date, would have potential for much more mileage than the already impressive increases over its predecessor.

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Dec. 12, 2000

# Chapter VI

## Oil Coolers

Why do people debate whether an oil cooler is necessary on air-cooled Big Twins and Sportsters for that matter?

### The Adversary is Heat

I just do not get it. There is no reason whatsoever, to not run one or even two as I do. Moreover, I am not talking about those cute little chrome ones either, although they are better than nothing is. I want those big, ugly to some, black coolers with ten-rows of turbulation enhancers, mixing the hot oil in order to obtain maximum heat exchange dissipation.

Heat is the enemy.

Why is there a Debate over Oil Coolers and Synthetic Oil?

It is like the synthetic oil versus fossil oil debate. Once an engine is broken in with fossil oil, there is no excuse for not switching to synthetic oil! Oh, the naysayers preach their half-truths passed down from the so-called experts. Differing opinions based on mythological mistruths bombard the Harley rider from every quarter.

What is he or she to do? Many times, nothing as they stay with the status quo for fear of damaging their rides.

It is this simple; let me say it again; heat is the enemy. The

proper oil cooler and the use of synthetic oil will allow your Twin Cam to run cooler. The use of synthetic oil will allow for easier low temperature weather startup. More importantly, oil will flow faster to starved and drained areas especially in the cold. Synthetic oil will also adhere to parts longer for startup after shutting the bike down. Furthermore, synthetics retain higher viscosity while subjected to high temperature, high stress conditions; oil that is more viscous is a superior protector of the engine.

Fossil oils break down at a faster rate from the ravaging effects of heat-oxidation than synthetics. Synthetics do not break down until much higher temperatures than fossil oils.

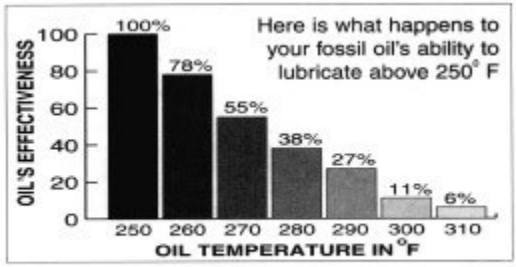
Finally, to counter a pervasive mistruth propagated by the fear mongers, there is no known case of bearing skid with the use of a synthetic. I have taken the trouble over the years to ask various bearing and oil companies that supply Harley-Davidson like the respected Timken.

This myth derives from the slipperiness characteristic of synthetic over fossil. I want slippery. I also want the superior film strength of oil that separates moving parts, sometimes only molecules thick.

Synthetic is tough; its film strength and slippery attributes protects the engine better than fossil oil.

The only downside I see is that synthetic oil is more expensive than its fossil counterpart is. Of course, the gap in pricing will become less as out-of-the-ground oil prices steadily rise.

A well-chosen oil cooler like Jagg and the use of synthetic oil will help protect against the great engine destroyer, ravaging detonation.




Detonation: The Great Engine Destroyer

Detonation defines as excess heat causing an unwelcome spontaneous complete ignition of unburned fuel anywhere in the combustion chamber in addition to the normal, planned, and timed spark ignition of the fuel mixture. Ignition, whether deliberately sparked or unprompted, causes a pressure flame front of burning fuel to travel across the cylinder.

Combustion Lag

Even though everything I describe happens in a mille-second, there is a physical principle called combustion lag. It

takes time for the fuel molecules to warm up, heat up, and then finally begin to burn. This combustion lag is at least partially responsible for advance/retard ignition timing.

Once burning the fuel mixture, the same lagging process moves across the combustion chamber. Mechanics call this traveling movement of igniting and burning fuel across the chamber a flame

Because of this time lapse, ignition must occur before the compression cycle finishes while the piston is still rising because it takes time for the fuel to get burning. The flame travel must complete its journey, as the piston is on its way down the power stroke to ensure the highest percentage of burn occurs for maximum expansion to create useable power.

There are no gas explosions since Octane prevents uninvited and unplanned outbursts of counter-productive activity…unless of course, detonation overwhelms the normal burning process.

One of the great advances in the early 1900's of engine technology was the discovery of Octane. There are many different Octane substances or elements, the best early example is lead, which control the rate of burn.

Before Octane use, the first engines hacked along with much inefficiency because of destructive and haphazard detonating explosions. Octane has the ability to stop chaotic explosions and bring order to the combustion chamber.

An email buddy, Mike always sets me straight when I am

wrong or sloppy in my definitions, “Octane is a very specific compound used for comparative testing of detonation resistance of other fuels. The “Chemical Soup “we call gasoline does not contain Octane and the additive we put in it does not add Octane. We add and delete components to enhance the Octane Rating. Tetraethyl Lead was the old and easy way, now it is a lot more complex. Some are even oxygenates (MTBE and all the alcohols) that also lean the mixture and diminish power unless we compensate for

Mike always wrecks my neat and simple explanations but I appreciate his correctness as it keeps me in line. Wait until the reader sees how he demos my detonation explanation.

Enhanced Octane reduces harm inside the engine but many times produces harm to the environment and to us. Lead, for example causes mental retardation and learning disabilities in children. It does so to a lesser extent with adults. The reason is that children will put substances in their mouths at one

time or another that adults will not. Kids will eat things like sweet tasting lead paint chips or suck on lead pencils.

Lead tastes good.

In the day, engine exhaust left lead residue everywhere on the ground and in the soil where children play. The government, not as fast as it should have, quite rightly banned lead after a half-century of exploited use. The problem is that lead is a wonderful albeit dirty Octane enhancer.

Successive Octane's did not wreak havoc with children's brains but they did cause cancer with all that exhaust permeating the air with silent debilitating and sometimes-carcinogenic killing fumes.

The advantage of the alcohols like ethanol is they burn cleanly. They are also a safe Octane enhancement.

However, Harley-Davidson needs to use resistant-rubber compounds in manifold seals and other induction components as ethanol will distort traditionally used rubber composites. Harley-Davidson warns against high ethanol levels in pump gasoline for this reason. Consult your manual for the acceptable ethanol level for your year and model. Gas pump signage usually has percentage levels of contained ethanol near the price per gallon or liter notice.

An Octane enhancer discourages unwarranted explosions. Octane causes fuel to burn at a predetermined rate.

How long it takes a flame front to travel across a combustion chamber with controlling Octane allows the engine designer and tuner alike to advance ignition timing or retard it. The spark can ignite the fuel at the most auspicious

time for maximum fuel burning known as the percentage of burn, thus creating maximum controlled heat. Rate of burn, piston speed, and cylinder diameter are some of the main factors determining ignition timing.

They also have a part in determining cam timing, when valves open and close. Thus the complex blueprint of valves opening and closing on previous and succeeding engine strokes; spark ignition occurring before a compressing piston reaches the top of its stroke…it is all about combustion lag, rate of burn, sizing of cylinders, mechanical parts movement lag, piston speed and on it goes.

## Heat is the Savior

The modern engine management system generates a very hot spark jumping across a large gap on the spark plug

electrode.

Compare the old mechanical points-condenser ignition in use up until circa-1977 with a spark plug gap of around .028” (twenty-eight thousandths of an inch), with a modern Twin Cam’s spark gap as much as .043” (forty-three thousandths of an inch). This represents about a 65% increase. In order to do this coil voltage increases from about 18000 volts up to and past 40,000 volts from the point’s ignition coil to the electronic one.

The engine needs this lean-burn air to fuel ratio in order to try to burn all of the gasoline and the residues generated from this process. Burning some of the generated residual oxides and other pollutants in this high-heat process is the goal from an emissions control point of view. Heat provides relief from the dictates of the Environmental Protection Agency (EPA), California Air Resources Board (CARB), and other agencies that monitor the quality of our environment through

regulation of emissions from internal combustion engines.

One major technique that manufacturers like H-D, engineer to comply is the use of heat to burn noxious pollutants.

Harley-Davidson cannot survive unless it is able to comply with ongoing decreasing emission directives.

## Heat is the Enemy

Excess heat is the enemy of Octane sometimes overriding it to produce detonation. Ever increasing EPA regulations dictate that as many emissions as possible incinerate inside the combustion chamber. This necessitates lean burning where the percentage of air is higher relative to fuel.

The high percentage of air molecules mixing with a smaller percentage of gasoline molecules is the cause of this excess heat. Lean burning creates excessive heat.

Modern ignition systems, or using more appropriate nomenclature, engine management systems have progressed a long way to assist the lean burning of fuel. At the same time, the system attempts to control the ravages of the great engine destroyer detonation.

Detonation also known as knock, spark knock and pinging sounds like a bunch of ball bearings bouncing around inside the cylinders. Ongoing detonation occurring rapidly with no predictability causes the pinging sounds that are so injurious to the engine.

I like many others used to think that detonation was the

violent collision of two or more flame fronts traveling across the combustion chamber. Others and I have written this misinformation so many times.

I was wrong.

My email buddy, Mike let me know this clearly, as he asked me rhetorically how I would explain dual plugging a cylinder head.

As soon as he asked this question, it cut right to the bone. Obviously, there would be a collision of flame fronts. Why is this not injurious as a fabricated detonation? Time to listen and learn or re-learn as is my case. After his devastating question, Mike went on to explain, “What actually happens is normal flame propagation generates a pressure wave as it travels across the combustion chamber.  If pressure and

temperature of the yet unburned mixture exceeds its ability to resist combustion the balance all goes off at once – detonates. There are many causes for detonation. The obvious two, poor quality fuel or over-advanced ignition timing are just the beginning. Anything that affects relative cylinder pressure is a part of the equation. Static compression ratio, valve timing, and the often-neglected rod to stroke ratio are very interesting variables to think about. When you increase the stroke with little or no increase in rod length, piston speed relative to degrees of crank rotation over TDC, go ballistic. Now consider that from the perspective of combustion chamber volume, pressures, fuel, and ignition requirements. Colliding flame fronts are an old wives tale." 2

Mike stomped me good but quite appropriately, on that one even though I corrected his grammar. That is everything except the end-of-sentence preposition. I am sure that when he reads this, he will also be updating my grammar.

Therefore, one flame front is as designed while detonation is a rogue spontaneous total explosion, out to create trouble. The rogue element disrupts the timing of engine components whose job it is to create power. If unwanted detonation ignites before or after the spark plug it will destructively burn fuel simultaneously at the wrong time so there is not enough fuel for burning when the legitimate ignition occurs. If it ignites at the same time elsewhere in the cylinder, it will grossly interfere with the legitimate flame front screwing that particular ignition cycle up.

Mike was kind enough to take the time to list some causal factors of temperatures reaching into spontaneous ignition ranges, which cause the counterproductive and ravaging detonation. Other unwelcome assists to detonation are lugging the engine, high rpm riding, excess summer temperatures, low-viscosity oil, and other mitigating factors.

# Lugging and Detonation

Lugging, which is running the engine with too low rpm in too high a gear has always been a problem for some.

Put another way, riding the bike at a too low rpm causes lugging whose symptoms include anemic power and lurching. The cure for lugging is always downshifting to a lower gear… sometimes two gears lower for the uninformed. It is a huge problem with the torquey 2007 and later TC96 with their supercharged primary gear ratios and high-ratio rear drive pulleys. The TC96 powertrains really cook when properly used out of the lugging rpm range but many H-D riders are having difficulty adjusting to these never-seen-before H-D gear ratios.

Lugging exacerbates detonation as well as amplifying the much complained about fifth gear chatter in the 2007 Cruise Drive 6-speed transmissions. The cure for both maladies, on my bike anyway, seems to be putting the engine under load in the proper gear at a sufficiently high rpm.

Of course, running two 10-row Jagg turbulated oil coolers also reduces cylinder head temperatures, thus controlling the possibility of detonation from the other end. However, we do not cool oil temperatures in order to relieve cylinder temperatures for offsetting lugging consequences.

A stock bike with stock compression only needs one turbulated cooler. Obviously ten rows of turbulation is superior to 8-rows, which is better than 6-rows and on it goes.

A high compression bike needs the two 10-row coolers. Running two oil coolers also necessitates using those that

have the least restriction to internal oil flow also known as pressure-drop.

Riding at a higher speed in each gear than what traditional riders have been accustomed is very necessary with the 2006 Dynas and all 2007 Twin Cams.

The keys are to avoid lugging; using synthetic oil once the engine is broken in and running a turbulated oil cooler with the least pressure drop…and then, of course we need to fool with the air fuel ration and ignition.

In Chapter III: H-D 110 Cubic Inch CVO, I examined the scourge of heat brought on by Stoichiometric air-fuel-ratios. I also gave the solutions to excess heat.

If transmission clatter-whine is an issue, either the BAKER

F6F helical gear replacement kit or the Harley-Davidson IDS rubber-stopped rear pulley kit should alleviate the noise.

Now, I would like to deal with a topic regarding the semantics of the words row and core. Jagg refers to the tubing in oil cooler as rows while other people in the industry and riders refer to it as cores. To be specific, and for our purposes, core generally refers to the cooling matrix of an oil cooler, which, comprise all the tubes, fins, and header plate together. Core sometimes refers to the whole oil cooler without specific mounting or specific application hardware. Use row or passage to refer to a single or multiple directional tubes as part of oil cooler.

H-D Response to Lugging

Harley-Davidson responds to customer's lugging the new Twin Cam 6-speeds in Service Bulletin M-1186 0n November

20, 2006, titled ECM CALIBRATION UPGRADE FOR 2007 BIG TWINS (PRODUCT PROGRAM 0903).

Lugging is a rider-induced engine problem exacerbated by the new supercharged gear ratios in combination with a 6-speed transmission and new riders who are overly cautious in riding their motorcycles. Therefore, they mistakenly ride in a lower rpm range in too high a gear thus overloading the engine. Lugging makes the engine work outside its comfort zone resulting in agonizing torque shock loads emanating from the engine and down the powertrain. H-D is particularly concerned with grabbing a handful of throttle under 1500 rpm. Well, I feel there is nothing wrong with hard acceleration in first gear under 1500 rpm with a warm engine that is broken in. However, if doing this from a stop or slow crawl in second gear, then the engine becomes overtaxed. Furthermore, doing this hard acceleration under 1500 rpm from a slow speed in third or higher gears will injure the

engine over time. Lugging can happen in any gear and at any conservative speed during low and mid rpm ranges with hard acceleration if not starting at a high enough speed. Slow acceleration in these lugging situations makes the situation a little better but not good and will still tax the engine.

As I have said there are not any highways in North America that will legally accommodate higher rpms in sixth gear. Most highway-riding easily accomplishes in fifth gear.

Harley attempts to address the problem of customer lugging with an ECM calibration upgrade that smoothes the torque curve in order to assuage high shock loading out the sprocket shaft and down the powertrain. This recalibration is already done on the production line after early November 2006. Production of next year models generally begins in July of the previous year. Therefore, any 2007 TC96, CVO 110 and

police specific Twin Cams built before early-November 2006 are eligible for this ECM recalibration upgrade. Your dealer or any H-D dealer for that matter can enter your motorcycle VIN (serial number) on www.h-dnet.com, which you the rider cannot access as it is a business tool for Harley-Davidson and Buell dealers to see if your motorcycle is affected. If so, the recalibration is free for as long as the bike is under H-D warranty.

H-D Gear Ratio Changes

The massive increase in the compensator engine sprocket from 25-teeth to 34-teeth and the clutch sprocket from 36–teeth to 46-teeth gives a primary ratio of 1.35 engine rotations for one rotation of the clutch hub and therefore one rotation of the attached transmission mainshaft that the hub sits on.

- This compares to the former primary ratio of 1.44 engine rotations for one rotation of the clutch hub and the transmission mainshaft.

- The final ratio for 1994 to 2006 (excepting the 06 Dyna) is 315:1, thus turning the engine 3.15 times for one rotation of the rear wheel.

- The ratio for the 2006 Dyna is 2.88:1 so that the engine turns 2.88 times per rotation of the rear wheel.

- Next, the final drive ratio for all 2007-2008 Big Twins is 2.78:1. Therefore, the engine turns 2.78 times for each rotation of the rear wheel.

The difference between the 2006 Dyna and all the 2007 and 2008 Twin Cams is a reduction of two-teeth on the rear

pulley of the 2007-2008 models to 66-teeth from the 2006 Dyna's 68-teeth, which also reduces from other 2006 and earlier Twin Cams more traditional 70-teeth.

The 2009 Twin Cams revert to the same final drive ratio as the 2006 Dyna. In 2009, the final drive pulley increases two-teeth/nubs to 68.

## USA Overall Gear Ratios

H-DI (H-D International) models generally have taller gearing for better highway touring because of increased speed limits. The taller gearing results mainly from smaller rear wheel pulleys than those used in USA. 2009 Touring models have different gear ratios than 2007-2008 all models. 2009 Softails and Dynas have the same gear ratios as 2007-2008 Twin Cams.

USA rear-wheel-sprocket-sizes are firstly the King of the Highway Shovelhead FLH Electraglide, which utilizes a chain drive final wheel sprocket of 51-teeth for the four-gear transmission, Seventy-nubs are standard for both the 5-gear transmission 1999 EFI and the 2005 Twin Cam FLHT belt drive final pulleys. The 2006 Dyna final drive pulley has 68-nubs and finally, 66-nubs for the final drive of all 2007-2008 Twin Cams. European final pulleys are as low as 61-nubs for high-speed riding on superbly designed and maintained Autobahn like highways. Overall gear ratios show the number of engine revolutions required to turn the rear wheel one 360-degree turn. For example in gear a 2008 model takes 9.31 engine revolutions to rotate the rear wheel once while a 1999 Big Twin takes 10.81 engine revolutions to obtain the same goal in first gear. Overall gear ratios take into account the internal transmission ratios with the primary and secondary gear ratios while the final drive ratio is a simple

multiplication of the primary ratio times the secondary ratio. I cover and explain these subjects in detail in Volume I: The Twin Cam in Chapter VII: The Controversial Harley 6-speed

## An Unpopular Theory

I have a theory that will offend many.

Some traditional riders are not used to these dramatic gear ratio increases so ride in a lower rpm range without thinking and realizing it because that is what they have always safely done. I know there is an offensive noise in many 2007 transmissions. I also know that aggravating this transmission noise along with engine detonation is lugging, which is riding in a gear at too low an rpm.

New riders many times lug older Harley's but are much

unprepared for the supercharged gear ratios beginning with the 2006 Dynas. However, the Dynas do not seem to present much of a problem because of their hot rod nature.

The 2007 to present Baggers are a much different story. I mean, whoever heard of breaking a new Hog in at up to 75 mph on the highway, which is what I did with my 2007 Streetglide with nary an adverse marking on the pistons. I know this because I hopped my FLHXI up after break-in and was anxious the see the pistons, which are still like new.

Detonation exacerbated by lugging occurs with rider ignorance of correctly matching rpm to the gear engaged sequentially increasing from low gear. In low or first gear, rpm must start at idle, which is around 1000 rpm. The rider should twist the throttle, accelerating away from a stopped position, steadily increasing the rpm through the power band. Neither baby nor abuse the throttle twisting in normal riding. Shift into the next gear near the top of the power band. If

shifting into the next gear at a low rpm, the bike will slowly surge, laboriously attempting to reach the comfort zone of the power band. Pinging, emanating from the cylinders (detonation), will lend noise to this inappropriate overworking of the engine.

Difficulty gaining speed is a major indicator of lugging. However, the difficulty in top gear may either be lugging or wind resistance. Speed will indicate which the case is. Going 50 mph in sixth gear is definitely lugging. Pushing the bike to 100 mph and over encounters much wind resistance.

Without benefit of a water jacket with a cooling radiator, that keeps engine temperatures within acceptable ranges, the poor Harley-Davidson air-cooled engines fall victim to the ravaging effects of detonation. While a water jacket is a cure for temperature control, it is not a Band-Aid to suppress the

effects of rider lugging.

## Heat is Pressure

The burning fuel creates heat. Heat expands creating pressure. This expanding pressure drives the piston down, spinning the flywheels and transferring this energy down the powertrain to the rubber where it meets the asphalt.

## Air and Liquid Cooled Engines

I hear all the time the ongoing controversy about whether oil coolers are necessary for a Big Twin or Sportster Harley-Davidson engine. Why there is not unanimous support universally for oil coolers is beyond me.

Back in 1999, I even heard an H-D engineer whom wishes to remain anonymous for obvious reasons say, that he thought the Factory did not recommend oil coolers for the new Twin Cams because the powers-to-be felt that the H-D rider would interpret their inclusion as an indication that there was something wrong with the new engine.

I can say, in all my discussions with Factory executives and engineers that they were extremely proud of their Twin Cam development accomplishment. They were also very protective of their new baby. Their legitimate fear as I interpreted was the traditional resistance of the Harley rider to change. Writers and mechanics like me would leap to Harley-Davidson's defense if criticized for using an oil cooler on an air-cooled engine.

Many bikes and cars do not need oil cooling because they already have temperature control. The engines are liquid cooled with a water jacket encasing the engine. Liquid cooling

keeps an engine's operating temperatures within a predetermined range of lower temperatures.

You may have seen on television or heard the stories of the V-Rod left running for a long time in a metal transport box in the Nevada desert with no air circulation whatsoever. The experience would have demolished an air-cooled engine like the Twin Cam of heat prostration in short order but the V-Rod did not miss a beat.

Why, because it is liquid cooled.

Oil Coolers are an Answer

My bike does not ping with 10.5:1 compression.

Well, that statement is not entirely true as it sometimes does if not aggressive off the

line. In some situations with cars all around, it is necessary to baby the Twin Cam.

I have always highly recommended using an oil cooler on a Harley except water-cooled ones. I strongly advocate their use on the very hot running Twin Cams.

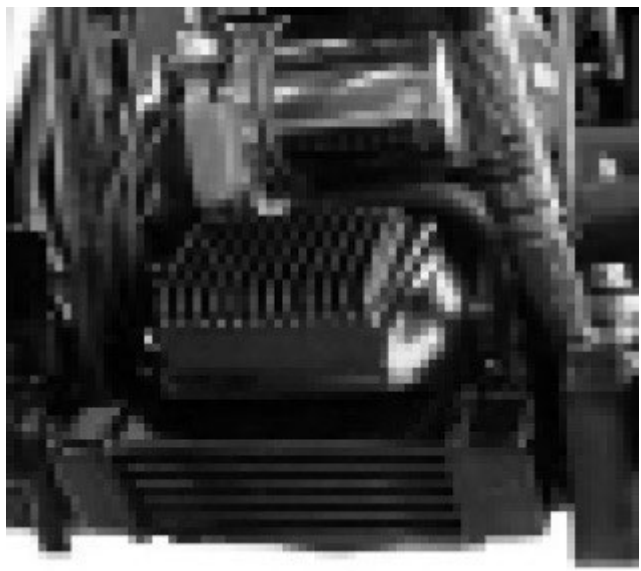
Now, the coolers I have used on the Twin Cams measurably do lower heat but I have never been happy with how much…that is, until last year in 2007. Bob Wood of Wood Performance who has a great knack for simplifying the complicated says to me that I need two 10-row Jagg coolers on my Twin Cam.

Huh?

Yes, you need 20-rows of Jagg oil cooling to lower the temperatures sufficiently in the heads to prevent detonation Well, well, well, is this not an interesting statement to install two of the biggest and most efficient oil coolers on my bike to control detonation. Bob says that head temperatures need reduction to between and maximum particularly in higher compression engines. 3 H-D allows for F with stock compression. In my opinion, most stock Twin Cams will suffer detonation at various times.

I followed Bob's advice and all I have to say is that it works and works well.

Low mount 6-row Jagg oil cooler



The 6-row 1270 Jagg Low Mount Oil Cooler, designed for late model EVO and TC88 and TC96 Dressers and Road Kings, is particularly suited for mounting on the Ultraglide with lowers. This compact unit offers up to three times the effective cooling surface of the closest competitor. The oil cooler

engineering design brings oil temperatures into a normal oil temperature operating range. The cooler's special inline turbulators and mini-louvered air fins insure unparalleled efficiency.  Note the compact and efficient design. A unique mounting system insures easy installation. The cooler interrupts the feed oil line to the engine. The inlet oil line feeds oil from the tank reservoir while the outlet oil line feeds oil into the engine via the necessary Twin Cam oil cooler adaptor. The chrome oil filter above the rectifier attaches to the oil cooler adaptor.

## Turbulation Balanced with Restriction is the Key

First, I will deal with turbulation, an oil-mixing technique to increase oil cooling.

Secondly, I will discuss a design phenomenon called oil pressure drop that we wish to avoid in our oil cooler

purchase, especially when running two big 10-row coolers in tandem.

The two concepts relate but are separate. However, both turbulation and pressure drop are very important; we want one but not the other.

How do you choose an appropriate cooler? For the most part, visual inspection can determine oil cooler efficiency. Observe the length of the cooling oil passages encased by finning, the number of fins, and the area of finning as a cooling surface, exposure to air, air temperature, and motorcycle speed.

The internal construction of the cooler passages and any restriction causing pressure drop is discernible by other means like sophisticated wind tunnel and bench flow analysis.

Some coolers have turbulation, which mixes the oil inside the cooling passages to allow maximum oil exposure to the cooling effects of the finning. Two oil coolers that I am aware of offer turbulation, Jagg and Harley-Davidson. Jagg used to manufacture the Harley-Davidson oil coolers until the Factory sourced them elsewhere. Read what you will into this situation but certainly Jagg was the first of the two to turbulate their oil coolers.

However, turbulation or mixing the oil is not a big secret nor is the process complicated. It is common sense derived from a logical thought process.

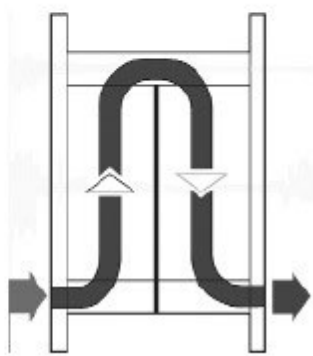
Laminar airflow into an induction throat is good as more volume moves in an orderly fashion. Turbulent intake air is not as good as it moves slower thus reducing flow volume.

The opposite principles are true for cooling oil. The goal is

to mix laminar flowing oil that only cools on the outside streams near the row surfaces by putting minor obstructions in the way of the oil flow. Engineers want to disrupt laminar flow in order to bring middle streams of hot oil in the center of the rows to the periphery where attached cooling fins can dissipate the transferring heat thus cooling the oil.

Jagg Turbulator, Finning, Two-pass Design and Oil Cooler

Adaptor



Top left, the picture depicts a turbulated row in a Jagg oil

cooler. Extensive finning to radiate heat exchange from the oil into the atmosphere would line all sides of this depiction of a turbulated row. For most of my mechanical life, I have made the misassumption that oil cooler rows that I incorrectly referred to, as cores are round and tubular. I guess some do have this design much to their cooling detriment. Some diagrams even depict the rows as round tubes even though their coolers use a flattened row profile. The goal is to mix or turbulate laminar flowing oil that only cools on the outside streams near the row surfaces by putting minor obstructions in the way of the oil flow. Turbulation disrupts laminar flow in order to bring middle streams of hot oil in the center of the rows to the periphery where attached cooling fins can dissipate the transferring heat thus cooling the oil. However, the very necessary turbulation inside the passages that is so vital to oil cooling and thus engine preservation does obstruct the oil somewhat. How much is what we need to know so we can choose a cooler with the least pressure Excess pressure drop may malnourish some areas of the engine with adequate

lubrication. Jagg was the first to turbulate their oil coolers. It also remains in the lead for the best combination of turbulation with the least pressure drop with near equal cooling properties.

Testing has demonstrated that the Jagg oil 10-row cooler has up to 3.5-times less pressure drop than the comparable Harley-Davidson oil cooler. Why would a rider choose an oil cooler with higher-pressure drop over those with less pressure drop like the Jagg, all other factors things being equal? Top right, the picture depicts Jagg finning outside the turbulated passageway. Bottom right, two-pass flow configuration, as related to oil cooler flow orientation describes a separator between the inlet and outlet, dividing the number of rows of the oil cooler into sections. The hot oil must enter and pass through the length of the cooling tube(s), make a 180-degree

turn, and make a second pass back the length of the cooling tube(s) thus making two passes. There are two major reasons oil cooler may have a two-pass configuration. The first is for plumbing configurations where the inlet/outlet location is important for space or design considerations. The second is performance. The performance reason requires a more in-depth explanation. Low gallons-per-minute (gpm), flow rate systems such as with Harley-Davidson applications respond well to a two-pass flow orientation in an oil cooler that offers low restriction. The reason is that it gives the cooling medium greater opportunity for heat exchange of oil-heat into the atmosphere by slowing the oil down just a little. However, two-pass flow orientation employs best when considered in the overall flow system. Contrarily, a cooler optimized for performance within a system with a single-pass flow configuration can gain a good deal of heat exchange by further slowing the oil's flow and creating a two-pass flow configuration but it will do so at the expense of significant pressure loss. Therefore, a two-pass flow orientation employs

best systematically within a total system utilizing turbulation without much pressure drop. Consideration of the total system is necessary as the best results derive from cooperation between differing factors of maximizing cooling and minimizing pressure restrictions. Finally, bottom left, Jagg's #4600 offset oil filter adaptor shown is necessary for mounting oil coolers on Twin Cams. The optional #4700 Offset Oil Filter Adapter with thermostat is also available for those wanting an automatic shut off-turn on valve set to F Either billet aluminum construction adaptor installs between the engine and oil filter for quick and easy access to oil supply for oil cooler installation. Moreover, the #4600 or the #4700 Offset Oil Filter Adapter is required for oil cooler installation on Twin-Cam bikes and is optional for Fuel injected Big Twin Evolutions. Many adaptor plates do not flow the oil through the oil filter first. The design of Jagg adaptor plates specifically flow oil through the oil filter before the oil cooler resulting in less restriction because the oil is warm and thus less viscous. This is important for the Twin Cam because of

the oil filter's placement on the pressure feed side of the oiling system. The Jagg adaptor offers less pressure drop than any other oil filter adaptor due to this purposeful directional flow characteristic. This flow path through the filter first, contributes to the total system of turbulation combining with less pressure drop that makes the Jagg cooler advantageous over competitors at this time of writing. The short version is that the correct volume of cool oil delivers to all the moving parts needing oil film strength to both separate and lubricate the mating components.

Resourceful Bikers

The resourceful biker of yesteryear would find a wrecked car and remove the transmission cooler. A few hose clamps and some extra oil line and voila´, perfectly functioning Harley-Davidson oil cooler. This methodology still works well

today for those inclined.

Today, we have the advantage of the Internet, where a little research can easily determine which ones, if any are turbulated.

What has the Factory advised over the years? Harley and many of its dealers has been saying forever that the Evo and now the Twin Cam 88 did not require oil cooler or synthetic oil for that matter. The Factory has begun to offer oil coolers as an option add-on for the last seven years or so. However, Harley did use non-turbulated Lockhart oil coolers (62571-77, black; 62577-77, chrome), from Lockhart Industries, Inc., formerly located in Paramount California.

- Lockhart oil coolers came on Big Twin Shovelhead sport models FX, FXS, FXEF, FXS-80, FXEF-80 1977 to 1981, (62551-82, black; 62559-82, chrome).

- They also came on the rubbermount 1982 FXRS and the FL, FLH, FLH-80 and the FLH Classic (62571-77A, black; 62577-77A, chrome), for a few years from 1978 to 1981.

In another part of this book, Chapter III: H-D 110 Cubic Inch CVO I discuss the fact that the Motor Company uses their Premium 8-row turbulated oil cooler (part number 26155-07) as a stock part on the FLHTCUSE with the thermostat activating at F C.).

H-D is having big problems with the CVO 110 from overheating. Rear head gaskets are blowing outwards from the oil return hole between cylinder and head in as little as 2000 miles of use. Harley must have had issues during development of the CVO 110 because the compression ratio is low at a paltry 9.3:1. It further lowers to between 9.15:1 and

9.2:1 in 2009. This is the antithesis to performance.

The stock TC96 Twin Cams have a respectable compression ratio of 9.2:1, which is okay for a stock model. The TC96 increases its ratio by .4 points from the TC88 ratio of 8.8:1.

In a perfect world, this engine needs a minimum ratio of 10:1 and preferably 10.5:1.

Reduced compression ratios will lower cylinder head temperature heat and resultant detonation.

The addition of an oil cooler to assist in lowering cylinder head temperatures will also control the devastating effects of ongoing detonation.

How Much Cooling is Necessary?

A good oil cooler will cool engine oil by at least 20-degrees Fahrenheit.

Cooling will obviously vary with differing conditions. No matter how hard I try, I cannot remember where I saw this cooling claim many years ago Oil cooler will lower oil temperature about 22-degrees Farhenheit from to F at a speed of 40mph in 77-degree weather. First, I would be very happy if oil temperatures stayed at F. I would be ecstatic at having temperatures as low as F.

Twin Cams do not run at these lower temperatures. Our goal is to lower cylinder head temperatures down to about F Detonation can control at this temperature but not at 230 to 250-degrees and beyond.

I am comfortable that the above quote is probably accurate for the precise test parameters mentioned but I would guess the figures apply to an Evo or even a Shovelhead using around a 5-row non-turbulated cooler and a richer air fuel mixture than the mandated EPA lean one of 14.7:1.

Now, I will get to my point. While descriptive and authoritative, the tested figures are for a specific engine; timed and tuned individually for that engine's and riders needs; with a specific air fuel mixture (AFR); probably for a carbureted engine while everything is precisely fuel injected today and with the cooler placed in one of a few possible locations with or without some air obstructions. I could go on but while informative in a general way, these test figures will not have any relevance to your bike except to demonstrate that an oil cooler, any type of oil cooler will lower oil temperatures somewhat to varying degrees dependent on many factors. I do not feel the quote has much relevance for the Twin Cam, which moves heat to the outside of the engine

via a hotter oil flow through internal oil-cooling jets directed underneath piston crowns.

Increased finning of 60% over the Evolution models does a very good job of dissipating the heat. However, the system easily overwhelms.

The courteous and knowledgeable Marv Beals, the public face of Jagg Oil Coolers will put the last nail in the coffin of this discussion, “We do not feel like it is correct to provide a specific temperature drop unless it can be considered as a product within an overall equation. For example, someone can say his or her oil cooler can lower temperatures by F and not provide supportive input parameters that show how they arrived at the performance conclusion…what if that supportive data was an outrageous inlet oil temperature of F and an ambient temperature of F? That would be a poorly

performing oil cooler that could only provide F drop under those circumstances…but the consumer believes it can do a great job because they do not have the whole picture. On a daily basis, I provide experts in such diverse fields as NASCAR engineers to Industrial Floor Sweeping Machine manufacturers with specific heat rejection data of our oil coolers. This same data provided to the consumer does not assist with product education…it is universally and unequivocally misinterpreted as how their specific cooler will perform in their specific environment….Jagg oil coolers are designed to allow your bike to maintain a standard temperature range." 4

Stoichiometric Air Fuel Ratio for Gasoline

Predecessor engines to the Twin Cam had less cooling fin area and contained more existing heat inside the engine. They

also ran a richer AFR, below today's closed loop EFI mandated Stoichiometric AFR of 14.7.

The EPA considers a Stoichiometric 14.7 parts air to one part fuel an optimum ratio for gasoline in which all air and fuel theoretically mix for complete combustion. Stoichiometric AFR is different for various fuels; Gasoline is 14.7; diesel is 14.6; Methanol is 6.4 while ethanol is 9.0.

How do we burn such a lean mixture of fuel? Heat is the short answer. Cylinder head design and mating piston dome shape assist mightily but the bottom line is that spark plug gap increases to allow a hotter spark to ignite the thinned out mixture. The hotter spark comes from a higher voltage coil.

Gasoline cools but there is not much in a Stoichiometric mixture.

Air heats and there is a lot there relative to the cooling properties of gas.

## Oil Cooler Thermostats

The first question is do you really need one? The reader can decide on what flavor suits their riding habits after this discussion.

In the old days, two elastic bands and a piece of cardboard were all we needed. We used to attach the piece of cardboard with elastic bands in front of the oil cooler to prevent the wind from robbing the oil of too much of its heat when the weather turned cold. The people with some loose change bought or made nice leather covers that did the same thing aesthetically.

Lockhart, which had been making oil coolers forever, was the first company that I can remember that supplied automatic thermostats as an option to an oil cooler purchase for a Harley-Davidson.

However, much as I tried, I could not find Lockhart's web address or a contact to interview. I asked Marv Beals from Jagg for help, "I am sorry to report the demise of Lockhart. It is always a sad day when we lose historically strong companies in our small market. I have always been of the opinion that competition helps to define us and helps to scratch out a market for all who are competing. I consider myself an amicable competitor and look forward to the shared market cultivation from all our efforts" 5

There are many other American made oil coolers on the market such as Hayden, (www.haydenauto.com), and Derale, (www.derale.com).

The early Lockhart thermostats were external technically good looking little gizmos that rerouted or bypassed the oil away from the cooler and back to the oil tank from the engine when the oil temperature drops below a predetermined temperature like, say, F. This temperature is a safe level, as we do not want to further and unnecessarily lower oil viscosity at operating temperatures. Many later thermostats became internal, incorporated into the oil cooler design as with the now defunct Lockhart 600 series.

Jagg incorporates a thermostat into its oil filter adaptor (part number 4700), for mounting a cooler on the Twin Cam.

There are manual bypass systems like the optional Jagg Bypass valve (part number 4000). The rider must remember to turn off this valve by hand whenever the ambient air

temperature drops below F. Do not confuse this temperature with the other noted figure of 180-degrees, which relates to the oil and not the air temperature.

If you do not ride in cold weather, a thermostat is not required. If you do, then one of the aforementioned techniques is necessary whether it is a simple cover, a manual bypass valve, or an automatic thermostat.

As stated earlier, turbulated coolers like Jagg and current Harley-Davidson coolers have impediments built into their internal channels or rows that prevent the oil from flowing through the finned tubes as a laminar stream.

Regular oil coolers just have finned straight-through tubing. Oil on the outside of the stream is able to dissipate heat but that in the center largely retains its high temperature defeating the purpose of a cooler.

Turbulation mixes the oil bringing maximum heat to the cooling heat dissipating fins. I think it obvious that turbulated oil cooler is the one to choose over a straight flow-through one as long as the pressure drop is not prohibitive to necessary oil

Oil coolers are in the business of dissipating heat. Therefore, the turbulated oil cooler with the most rows or passages will be the most efficient. Next, we will choose the turbulated oil cooler with the most rows that has the greatest area of finning. Finally, we will choose a black one over beautiful polished billet or lustrous chrome because function and not form is what we care most about with performance components.

The next area that will be a major determining factor in

our choice will be pressure

# Oil Cooler Pressure Drop

Pressure drop refers to restriction in the oil cooler that causes oil pressure to drop below its normal flow characteristics.

## Definition of Pressure Drop

Pressure drop is the measure of pressure at the oil cooler inlet less the measure of pressure at the outlet of the oil cooler. Pressure drop will usually quantify for our purposes in pounds per square inch (psi).

## Definition of Heat Rejection

Heat rejection defines as a quantifiable measure of the rejection of waste heat from the engine into the atmosphere. Heat rejection is very similar but not synonymous with heat dissipation or dispersion away from the engine into surrounding air. Heat exchange is another popular term. There are nuances to each not readily apparent to the novice.

The term heat rejection conveys a better description of the goal of motorcycle oil cooler. For example, heat rejection is often the goal of a heat exchanger but not always. A car's heater core primary function is to exchange heat from the heater core into the car's interior seating area where its benefits can offset the cold. On a Harley-Davidson, heat exchanges or rejects from the cooler into the atmosphere.

Heat dissipation has more of a subtle difference from rejection and describes more of an inactive dynamic such as

passive radiation from a heat sink.

Now that Marv and I have dispensed with the semantics and the fine lines between terminologies, in the everyday world most people take poetic license with the various terms and interchange them as I have done in much of this article before I knew better. I can only go back in my mind to say that the term heat rejection was not descriptive of what I thought was going on, namely dissipation. After going through all of this, I am now comfortable with the term heat rejection; as well, the reader may be also.

Pressure Drop

The turbulation inside the passages that is so vital to oil cooling and thus engine preservation does obstruct the oil somewhat. I am going to qualify what is about to follow.

The Wind Tunnel and Flow Bench Analysis graphs demonstrates that the 10-row Jagg oil cooler is much more free flowing than its counterpart 8-row Harley-Davidson one. I think it important to say that on a stock H-D ridden normally the increase in pressure drop on a single H-D cooler will not affect the bike's engine longevity in a serious way.

However, from a performance perspective, I cannot say that. With this said, why wouldn't a rider choose an oil cooler with less pressure drop like the Jagg over those with higher pressure drop, all other factors things being equal?

Jagg has conducted testing that demonstrates that cooling properties of both Jagg and H-D oil coolers are equal. However, the Jagg oil cooler has 3.5-times less pressure drop than the comparable Harley-Davidson oil cooler. 6

According to the following published analysis by Jagg, “SUMMARY: Wind Tunnel and Bench Flow Analysis PURPOSE STATEMENT: To compare the important performance aspects of the Jagg FLH Oil Cooler with its closest competitor, the Harley-Davidson FLH Oil Cooler… (Comparison between the) H-D FLH 8-row cooler and the Jagg FLH 10-row cooler.

Notably, the Jagg Oil Cooler achieves maximum cooling effectiveness with up to 3 ½ times less pressure drop. At practical oil flow, volume the performance of both subjects is virtually identical. More simply stated; it does not matter how well the oil cooler performs if it restricts adequate delivery of the oil to the intended areas of the motor. The Jagg Oil Cooler bests the competition…again. It is important to note that this is a comparison of components. Future comparisons will detail the additional performance and flow advantage of the Jagg oil filter adapter’s oil flow theory and its effect on system pressure.” 7

The reader may have noticed that the Harley-Davidson oil cooler has 8-rows or passages while the Jagg one used for comparative analysis has ten. Furthermore, Jagg's literature states that the cooling properties are equal. This begs the question; does this not mean the H-D oil cooler is superior because it accomplishes the same cooling job with 20% fewer rows than Jagg?

I put this to Marv Beals.

He responds, "Without the entire picture, it could be misinterpreted to mean the 8-row cooler has better heat rejection characteristics. However, this statement and your question point directly at the heart of my entire focus…you see, if the other products of our heat rejection dialog are not considered, i.e. pressure-drop, then the aim of the entire

discussion is missed. The aim of the discussion is that in order to be fair and correct, one must consider the entire system. In other words, we could alter our turbulators to hold the oil in the cooler longer (create more restriction) thereby giving the ambient air that is blowing across the outside of the oil cooler a greater chance to reject the heat but we choose to balance heat rejection with flow. Given that both oil coolers are intended to be installed on the pressure side of the oiling system we can then state how significant it is that we are providing our heat rejection while still allowing the oil to get where the flow and pressure intend it to get to. In other words, it doesn't matter how cool the other guys can get the oil (and they don't get it any cooler than we do, mind you) if this cool oil is not getting where it needs to get to in order to do its traditional cooling and lubricating duties." 8

In short, oil-cooling properties are irrelevant without including pressure drop analysis. We can increase cooling efficiency with fewer rows by restricting the oil and keeping it

in the oil cooler passageways for a longer time-period.

On the surface, this sounds great. However, we are also restricting the oil's primary function of lubricating by retarding it's travel time to get to these necessary parts that require the oil's film strength to separate moving parts… cooler oil not getting to where it is needed does not help parts that are starving for maximum lubrication.

## Practical Tips: Oil Cooler Installation

I cannot tell you the number of dented front fenders we have repaired over the years because of telescoping front ends compressing into a recently installed oil cooler mounted sideways on the upper front down tube frame members horizontal with the ground. Always, compress the front end to see where oil cooler location is horizontally permissible on front frame down tubes.

Jagg coolers do not face this problem as they mount near the lower front motor mount or on the side of frame down tube. Locating an oil cooler down by the front motor mounts can be risky and downright dangerous below the frame. Hitting a curb or other obstruction can dump all your engine oil onto the ground. Unnoticed, this will demo your engine from lack of lubrication or worse cause a slippery accident from an oil soaked tire.

Of course, manufacturers like Jagg and H-D are very aware of this. Their coolers come with bracketry that keep their coolers out of harm's way.

If mounting an oil cooler that does not have this safety feature, be aware that a lower frame member must protect the cooler. Furthermore, oil lines must locate away from other

parts that may abrasively wear through them over time.

Do not locate an oil cooler in an area with obstructed airflow. Unbelievable as it sounds, I have found oil coolers inside fairings and behind lower fairings. Reduced airflow equals reduced cooling.

Important: Return or Feed Mounting

On pre-Twin Cam Harley's, oil filters after the return side of the oil pump. Pre-2000, external gear-driven oil pump, the oil cooler should interrupt the return oil line from the engine to the oil tank. The return line from the oil pump feeds into one side of the oil cooler. A second line exits from the other side of the oil cooler to the return fitting on the oil tank.

The Twin Cam is the first Harley to filter oil on the oil feed from the oil tank to the engine. The engine-feed line from the

oil tank supplies oil into one side of the cooler.

The majority of oil coolers are generic in that it does not matter which nipple is the feed into the cooler and which one is the return out of the oil cooler. On a few other oil coolers, there is directional oil flow with a feed and return nipple on the cooler that should be marked as such for the observant mechanic. Always read the instructions. Always check the cooler for directional intake and outflow indicators.

The vast majority of oil coolers including Jagg have no “In” and “Out” on the oil coolers. Whatever port the oil enters, it will exit the other one. However, some adaptor plates for mounting them may. If so, they will be clearly marked and again the installation instructions will indicate this if necessary

There may also be helpful suggestions for oil line routing for ease of fitment and installation.

The design of Jagg adaptor plates specifically flow oil through the oil filter before the oil cooler. Less restriction occurs because the oil is warm and thus less viscous. This is significant for the Twin Cam because of the oil filter's placement on the pressure feed side of the oiling system. Therefore, the Jagg adaptor offers less pressure drop than any other oil filter adaptor due to this purposeful directional flow characteristic. This flow path through the filter first, contributes to the total system of turbulation combining with less pressure drop that makes the Jagg cooler advantageous over competitors at this time of writing.

Concisely, first hot oil and then cool oil delivers efficiently to where it needs to go.

# Types of Jagg Oil Coolers

There are three basic types of vertical mount oil coolers for Harley-Davidson applications.

The first is the Slimline (part numbers 1200 and 1250), oil cooler. Smooth end caps help this cooler blend in to the frame and hardware for a clean, streamlined look. The cooler mounts to the left frame down tube, spaced away from the frame to allow airflow to both the oil cooler and engine's cylinders. The cooler mounts to the frame using unique, highly durable, Uralite mounts, which have a shock absorbing effect and clean styling. The Slimline's six-tube cooler with a two-pass design offers three times the cooling surface than Jagg's closest competitor…with special in-line turbulators.

The second is the Deluxe Oil Cooler (part numbers 1000

and 1050). The Deluxe cooler also, mounts to the left frame down tube. It too spaces away from the frame for superior cooling to both the cooler and the engine. The cooler also uses Uralite mounts to the frame. The Deluxe Oil Cooler's, six-tube, two-pass, design again offers three times the cooling surface than their closest competitor. Special in-line turbulators add to the cooler's efficiency with minimal oil pressure loss or The aluminum construction has a durable powder coated finish. Billet aluminum end caps with machined fins match and complement the style of the engines cooling fins.

The third cooler (part numbers 2100 black and 2180 chrome for Softail 2000 up and 2200 black and 2280 chrome for FLH 1999 up) is my favorite and is of course, the 10-row oil cooler.

I prefer black in the case of oil cooler. Chrome is a heat

transfer barrier that will hold more heat in than bare metal. Polishing is okay as it is not a barrier to heat transfer. However, it is tough as in real to polish a cooler without disturbing the fins and wreaking havoc with the heat transfer components. Now, it is true that black powder coating is also a heat transfer barrier but nothing like the chromium.

Of course, the parts of the oil cooler that is actually responsible for exchanging the heat have only a negligible coating on the leading and trailing edges of the fins. Powder coat is extremely durable, long lasting and is the best choice in my opinion for a functional oil cooler protective covering.

Jagg's coolers are all available in black. Some have chrome accenting and others have removable chrome accents. The Deluxe coolers are available in black or chrome. On chrome Jagg coolers, only the outside framework is chromed. Jagg does not saddle the rows and fins with this heat barrier because heat exchange function is much more important than

a mere chrome bauble. The available chrome accents on some Jagg coolers fit on the sides but again not on any functioning component.

They are also a separate purchase item (part number CH-160), to dress up some models of the black coolers. They require heat resistant silicone adhesive to attach to the sides of the 1200 and 1250 Slimline coolers, the 1270 FLHT low mount coolers and the 2300 FLH Twin Cam 10-row low mount oil coolers.

The vertical 10-row coolers mount on the left down tube frame member like the Slimline and the Deluxe coolers except it alone mounts parallel with the airflow and not against it as the other Jagg coolers. The in-line mounting orientation insures that the cooler takes up as little space as possible while the unique airfoil creates a low-pressure zone, drawing

air inside out through the oil cooler. With the air rushing by the oil cooler in an area of relatively high pressure, the unique air foil simultaneously creates a low pressure area behind the air foil and draws air through the cooler from its backside; the greater the speed of the air passing the foil, the greater the volume of cooling air drawn through the cooler.

The turbulated 10-Row oil cooler's two-pass design offers significantly more cooling surface than other H-D design oil coolers, again with the all-important minimal oil pressure loss or

Jagg's horizontal oil coolers mount near the front motor mounts but above the lowest frame member behind the front fender. The 2300 series FLH Twin Cam low mount 10-row coolers do not have an attached airfoil since they face directly into the air stream. Also available are 1270 and 1280 series 6-row FLH Twin Cam low mount coolers as well as the horizontal 6-row 1290 series for Softails. Lastly, Jagg sells a

universal mount (part number 3080), turbulated mega 14-row oil cooler that measures 8.25” x 5.25” x 1.5”

The Jagg thermostat actuator begins to move at F, and by F it fully-actuates, plugging the by-pass hole, and directing 100-percent of the oil’s flow to the oil cooler.

Oil coolers also add some oil capacity to a Harley-Davidson. A 6-row oil cooler will add approximately 6oz. of oil capacity while the 10-row oil cooler will add approximately 8oz. capacity. 9

More oil is good. I highly recommend a turbulated oil cooler with multiple rows, ten or more being preferable and with minimal pressure drop on all Harley-Davidson’s specifically those manufactured by Jagg. I feel that they are a prerequisite for any stock application.

They are an absolute must for any performancing options. Again, I believe in running two-in-line 10-row Jagg coolers on performanced Harley's with higher compression ratios. I do this on my personal bikes.

Oil coolers just make sense and pay for themselves by decreasing the incidence of detonation and by prolonging repair intervals.

Two Pass Design

I asked exactly what a two-pass design entailed and what it means in the oil cooler heat exchange world.

Marv replies, "Two-pass flow configuration, as related to

radiator and oil cooler flow orientation, means that there is a separator between the inlet and outlet, dividing the number of rows of the oil cooler into sections. The cooling fluid must enter and pass through the length of the cooling tube(s), make a 180-degree turn, and make a second (two) pass back the length of the cooling tube(s). There are a couple of reasons an oil cooler may be produced in a two-pass configuration; inlet/ outlet location for plumbing requirements and performance. Plumbing requirements are simply where you need the inlet and outlet ports to be located in order to plumb the oil cooler. The performance reason is a bit more in depth. Low gallons-per-minute flow rate systems such as with motorcycle applications respond well to a two-pass flow orientation in oil cooler that offers low restriction. Why, because it gives the cooling medium greater opportunity for heat exchange by slowing the oil down just a bit. However, two-pass flow orientation is best employed when considered in the overall flow system. Contrarily, a cooler optimized for performance within a system with a single-pass flow configuration can gain

a good deal of heat exchange by further slowing the oil's flow and creating a two-pass flow configuration but it will do so at the expense of significant pressure loss. Therefore, a two-pass flow orientation is best considered and employed systematically." 10

So here, we go again with a theme that just keeps cropping up every time I try to separate the different functions of oil cooler. If we want to understand how a cooler works, we have to look at everything put together as a system and how each role affects other roles.

A non-turbulated cooler can move oil through faster and with less restriction. However, the laminar flow only cools oil on the outside layers leaving central layers un-cooled and retaining heat. The name of the game is maximum-cooling, balanced with as little restriction as possible.

Turbulation or mixing of the oil offers maximum cooling but causes some restriction. Jagg is able to balance the two better than other coolers to offer less restriction than competitor coolers. The two-pass system offers much to the systems approach to provide the optimal oil cooler for a Harley-Davidson.

It may seem like I am slagging the H-D oil cooler. I am not as it is a good one. It is my second choice for a single cooler. However, I would not combine two of them in sequence because of the combined pressure drop values leading, in my opinion to too much flow restriction.

Mounting two 10-Row Coolers

The Jagg 10-row vertical oil cooler design mounts on the left side of a Harley on the front frame down tube.

Mounting a second one on the right side presents some logistical problems. The bracket welded to the cooler angles to keep the cooler in-line with the bike to save room rather than having the cooler stick straight out sideways.

The 6-row vertical coolers do stick straight out but there are no interference problems because they are less obtrusive than the 10-row coolers.

We could mount the cooler upside down, which would not affect cooling capabilities but the hose ends wind up on top instead of the bottom. Therefore, a plumbing confusion results unless the owner likes hoses traveling in many directions. In the wacky world of H-D, sometimes-mass complexity can be beautiful. I tried to mount them one on top of the other on the left front down tube but did not have quite enough room. This

would look cool, giving an all-business tough appearance. Bending the bracket carefully after heating it would do the trick for right frame-tube mounting.

I asked Marv Beals at Jagg if there was a front and back to their oil coolers. I was sure that was not an issue but it never hurts to ask.

He responded, “Hi Donny, you are correct, there’s no front and back. The problem is with the direction of the bend of the airfoil in relation to the mounting angle of the cooler on the down tube. Since these (10-row oil cooler) are designed to mount in-line behind the airfoil on the left down tube, if you attempt to mount the airfoil up-side down on the right side the angle will be off.” 11

The bracket could also be carefully cut-off and re-welded at the correct angle to mount the otherwise left handed cooler

for a right side mounting application.

Vertical 10-row Jagg Oil Cooler

The vertical 10-row Jagg oil cooler mounts to the left frame down-tube spaced away from the frame to allow airflow to both the oil cooler and engines cylinders. The cooler mounts

to the frame using highly durable, Uralite mounts which have a shock absorbing effect and clean styling. The in-line mounting orientation insures that the cooler takes up as little space as possible while the unique airfoil creates a low-pressure zone, drawing air inside out through the oil cooler. The 10-row oil cooler's two-pass design offers significantly more cooling surface than comparable coolers on the market. Special in-line turbulators add to the cooler's efficiency with minimal oil pressure loss or restriction. The construction is aluminum with a durable powder coated finish. Turbulators mix the oil internally in the rows so that all the oil cools. Non-turbulated coolers only cool the laminar flow on the outside near the passage walls leaving center laminar flow hot. Jagg oil coolers have less pressure drop or restrictiveness than H-D coolers.

## Negative Vacuum Principle

The 10-row cooler is considered by some as too wide to be

mounted straight out from the frame member directly into the wind. It does not bother me because I have a front crash guard on my Streetglide. However, it would not offend me either if mounting on a Dyna without a crash guard but ugly and beauty is in the eye of the beholder.

Jagg decided to conserve space because of the width of the 10-row cooler so they put an angled airfoil at about 45-degrees from the face of the cooler. Now the cooler can be run back in-line with the down frame tube. The cooling fins are NOT facing the air stream but the cooler is running parallel with the airflow and the fins are perpendicular with the air. However, the mounting has no obstructions in front of it allowing the air stream to flow right by without hindrance. I mention this because other oil cooler mounting locations partially hinder direct contact with the air stream. An example is those oil coolers in a horizontal lower frame mount position with a fender and wheel in front.

My physics education tells me that this will work very well, perhaps not quite as well as if directly in the wind but I was a little uncomfortable with it, nevertheless. The airfoil creates a vacuum between it and the outside (left) side of the cooler. The negative vacuum sucks air through from the inside or right side of the cooler. The reassuring part is that I can stick my hand down between the airfoil and cooler whenever the urge strikes to feel if this works in practice. All summer, I did this at different speeds and temperatures. It works great. The hot dissipating air coming off the cooling fins was testament to this fact.

Direct Mounting the Second Cooler

There is a second minor situation for right-side mounting and that involves the exhaust pipe partially being in the way. To resolve this, I simply angled the Jagg Ultralite frame

mounting clamps by swiveling them forward a little to angle the cooler out past the front pipe. Of course, I removed the airfoil because it would no longer be at the right angle for negative vacuum to be effective. However, the rows and fins are in the These frame-mounts help space the cooler away from the frame on both sides to allow airflow to both cool the oil coolers and the engine's cylinders. What would be the point of obstructing airflow to the engine when we are trying to cool engine temperature down?

I guess some people will think two big black oil coolers on one bike will not look right. I can tell you that I have to point out to most riders that there are two coolers on my bike, as they do not notice.

All I can say is that the dual 10-row Jagg coolers work and they work well. If Bob Wood's idea catches fire with the

performance set and/or as a way to deal with detonation on stock bikes then I am sure Jagg would manufacture left and right 10-row coolers.

I have said before but it bears repeating, I do not have a detonation problem on my 2007 FLHXI even though I have increased compression to 10.5:1 and ride aggressively in hot weather.

Conclusion

I would like to make two points here. The first is that the 10-row Jagg cooler has less potential restriction than an 8-row Harley-Davidson cooler because more rows offer less restriction as there is more area for the oil to expand into but the pressure drop on the Jagg cooler is still 3.5 times less than that of Harley-Davidson. This is a huge differential.

The second point is that if using a high performance oil pump like Feuling (www.feulingparts.com), or Delkron (www.delkron-mfg.com), pressure drop becomes an even greater concern. Therefore, in my opinion the Jagg coolers, which have substantially less pressure drop than the H-D coolers now become the oil cooler of choice.

Adequate oil cooling slows down the destructive oxidation forces that cause oil deterioration. Detonation, the great engine destroyer can tame by adequate oil cooling all other factors being in tune.

Finally, I think that every rider of an air-cooled Harley-Davidson should consider running an oil cooler on their bike. Yes, Harley's have been running well for a long time without the benefit of cooling but they have never run hotter because of dictated EPA Stoichiometric Air-Fuel Ratios.

I am going to climb out on a limb here; a sturdy one I might add, and state that I feel we are the last generation to ride air-cooled Harley-Davidson V-Twins.

Oil coolers are one factor of many that will prolong the lives of our beloved air-cooled dinosaurs.

## Oil Pressure Drop, Wind, and Bench Flow Analysis

### Preamble

Let me first expand the acronyms, some obvious and some not; gpm = gallons per ft/s = feet per btu/h = British Thermal Units per hour and 1.0 bar (barometric pressure) = 14.58 psi (pounds per square inch). The inner flow rate of 0.793 = 0.793 gallons per minute of oil the inner flow rate 1.321 = 1.321 gallons per minute of oil flow and the inner

flow rate 2.642 = 2.642 gallons per minute of oil

The following charts demonstrate the measured critical output of both H-D and the Jagg FLH oil coolers using the same input parameters. Deciphering the acronyms; TTI = Oil Temperature In; TTO = Oil Temperature Out and TSI = Ambient Temperature.

The following charts demonstrate the measured critical output of each oil cooler using the same input parameters.

Wind Tunnel and Bench Flow Analysis

Pressure Drop

Entry Pressure: Inner Circuit:

1.0 bar (barometric pressure of 14.58 pounds per square inch)

H-D FLH 8-row Oil Cooler Jagg FLH 10-row Oil Cooler

RESULT: At practical oil flow volume, the Heat Rejection performance tables below of both subjects are very close. More simply stated it does not matter how well the oil cooler performs if it restricts adequate delivery of the oil to the intended areas of the motor. Notably, the Jagg oil cooler achieves maximum cooling effectiveness with up to 3 ½ times less pressure drop. 12 Compare the underlined above example Pressure Drop of an H-D oil cooler of 4.714 psi to the same condition Jagg oil cooler drop of only 1.900 psi at 1.321 gallons per minute oil flow entering the coolers at F. Similar pressure drops were experienced at 0.793 gallons per minute,

2.619 psi vs. 1.100 psi at F and at 2.642 gallons per minute,

11.169 psi vs. 3.200 psi again at F.

Heat Rejection (Dissipation/Exchange)

Inner flow rate @ 0.793 gpm

Inner flow rate @ 0.793 gpm

H-D FLH 8-row Oil Cooler

Jagg FLH 10-row Oil Cooler

Heat Rejection (Dissipation/Exchange)

Inner flow rate @ 1.321 gpm

Inner flow rate @ 1.321 gpm

H-D FLH 8-row Oil Cooler

Jagg FLH 10-row Oil Cooler

Heat Rejection (Dissipation)

Inner flow rate @ 2.642 gpm

Inner flow rate @ 2.642 gpm

H-D FLH 8-row Oil Cooler

Jagg FLH 10-row Oil Cooler

I asked Marv why three examples of Heat Rejection tables were given?

Marv replied, will be as many heat rejection results/ comparisons as there are composites/sets of input variables." 14 Actually, Jagg tied the analysis together very well. If the reader checks out the three different gallon per minute (gpm), oil flow rate variables given in the first table of Pressure Drop of 0.793 gpm, 1.321 gpm, and 2.642 gpm, Jagg does a Heat Rejection Analysis of each of the three variables in the three successive tables to tie the Pressure Drop and Heat Rejection Analysis together.

RESULT: At practical oil flow volume, the Heat Rejection performance tables above of both subjects are very close. However, it does not matter how well the oil cooler performs if it restricts adequate delivery of the oil to the intended areas

of the motor. Notably, the Jagg oil cooler achieves maximum and near equal cooling effectiveness with the H-D cooler but most importantly the Jagg cooler has up to 3 ½ times less pressure drop than the Harley-Davidson oil cooler.

## Jagg Oil Cooler Installation Instructions

### IMPORTANT INFORMATION

THIS JAGG OIL COOLER MUST BE INSTALLED FOLLOWING THESE INSTRUCTIONS. READ THE EASY TO FOLLOW INSTRUCTIONS FULLY PRIOR TO STARTING THE INSTALLATION OF THE OIL COOLER KIT. CORRECT INSTALLATION IS THE ONLY WAY TO INSURE PROPER OPERATION OF THE OIL COOLER KIT.

### Basic System Installation Guidelines

Twin Cam models require the Jagg Oil Filter Adaptor #4600 or Oil Filter Adaptor with built in automatic thermostat #4700, which sells separately to access the oil supply.

To install the 4600 or 4700 adaptors:

1. Remove the oil filter and install the O-ring half of oil filter adaptor.

2. Then install the hose fitting half of the oil filter adaptor.

3. Install the oil lines.

4. Reinstall the oil filter.

5. Hook up the oil lines to the oil cooler.

6. Allow the bike to heat up to operating temperatures.

7. Finally, shut the bike down and allow cooling.

8. Retighten the Allen head mounting screws.

Instructions that are more detailed come with the adaptor. Novice mechanics must read these and it sure does not hurt for the rest of us either.

On rubbermount, Twin Cams allow adequate clearance to compensate for the motor shaking.

Oil Flow Direction

The oil flow direction from the adaptor is as follows;

1. Oil feeds the cooler from the fitting on the left side as viewed from the front of the hose-fitting half.

2. Oil returns to the fitting on the right side as viewed from the front side of the hose-fitting half.

General Instructions

1. Route the oil hose to avoid any hot surfaces or moving parts.

2. Insure all bends are smooth with no sharp turns that may restrict oil supply to the engine.

3. The oil cooler design is to mount according to these instructions. Any modifications may lead to a decrease in performance or item failure.

4. Always use a sharp knife or single edge razor blade when cutting oil hoses. Make a straight clean cut and at to the oil hose. This will insure a proper fit where the oil hose attaches to its connection.

5. The oil cooler mounting clamps included in this kit are designed to fit the frame tube diameters specific to the applications listed for this kit #. Any alterations or modifications to these clamps may cause failure of the clamp. Over tightening hose-clamps may cause oil leaks.

CAUTION: ALLOW MOTORCYCLE TO COOL BEFORE ATTEMPTING INSTALLATION OR RISK SERIOUS INJURY.

# INSTALLATION INSTRUCTIONS

## General Vertical Oil Cooler Mounting

1. Determine cooler placement on left frame tube by holding cooler to approximate height of desired location.

2. Choose the appropriate size clamps for your application according to specific frame diameter at the point of mounting. For 1000/1200, XL kits use included rubber shim material with 1 ⅛” clamps for a 1” frame diameter.

3. Spread clamps and place around left frame tube.

4. Position the oil cooler vertically on the left frame tube

with oil inlet and outlet pointing downward.

5. Align the boltholes in the oil cooler with the holes in the mounting clamps and install the nuts-and-bolts.

6. Carefully slide cooler and clamps up or down to position of desired height. Keep clamps free from any braces or indents that may interfere with a clean fitment. Insure the cooler mounts at a (straight out) from the bike to allow clear airflow in the case of Slimline or Deluxe Oil Coolers. Ten-row coolers mount straight back with airfoil to the outside and inlet/ outlet fittings facing down.

7. Tighten bolts. The cooler should firmly mount now.

Note: If installing the oil cooler using an oil filter adapter, the adapter is your oil supply source; refer to the instructions supplied with that item. Twin Cams require an adaptor #4600

or an adaptor with thermostat #4700.

On Evo and Earlier Big Twins NOT Twin Cams

1. Locate the source of the return line to the oil tank on your bike. This may be on the oil pump or oil filter depending on the model. Refer to illustration by model on instruction page number two.

2. Place a clean oil pan under the point of the return line fitting. Disconnect the oil return line from the fitting at its source. Some oil will drain out when disconnected. This oil can be reused if it remains clean.

3. Place a new hose clamp onto one end of the new oil hose and install onto the return hose fitting. Tighten the hose clamp securely.

4. Run this oil hose forward to the oil cooler where it is to attach to one of the fittings. Determine the correct length and cut the hose.

5. Place a new hose clamp onto this end of the hose and install onto cooler. Tighten the hose clamp securely.

6. Place a hose clamp onto one end of the remaining piece of new hose and install onto the other fitting on the oil cooler. Tighten the hose clamp securely.

7. Run this hose back to the return line source. This end of the hose splices into the existing oil return hose that was disconnected in step 2.

8. If the end of the existing oil return hose disconnected in step 2 is hardened or starting to deteriorate, trim off the bad

portion of the hose until you reach a good section of hose.

9. Determine the correct length of hose and cut the hose.

10. Place a new hose clamp onto the loose end of the oil return hose and insert it onto the other end of the black nylon hose mender supplied in kit. Tighten hose clamp securely.

On All Model Installations: XL, Big Twin, and Twin Cams

1. Inspect the oil hoses to insure there are no tight bends that may restrict oil flow and that they are not contacting any moving parts. If necessary, secure the new oil hoses to the frame with plastic zip-ties.

2. Refill the engine with the correct amount and type of oil. Check the level with the dipstick.

3. Start the engine and let it idle. Check all oil hose connections for any leakage. Tighten any hose clamps that may be leaking. Note: Over tightened hose-clamps may cut into oil lines and cause subsequent oil leaks.

4. After engine has warmed, feel the oil cooler. It should be warm from the hot engine oil flowing through it. If the engine is warm but the oil cooler is not, the oil is not flowing correctly or not flowing through the cooler at all. Retrace oil routing instructions to solve this problem immediately.

5. After installation, completion, and engine warm up; shut the engine down and recheck the oil level. Correct the oil level if necessary; do not overfill.

6. On Twin Cams shut the bike down and allow cooling.

Finally retighten the Allen head mounting screws on the adaptor.

FLH Twin Cam General Horizontal Oil Cooler Mounting

(Softail mounting is essentially the same but slightly different. Always read and follow the instructions provided or access them from (www.jagg.com).

1. Remove oil filter and install Jagg oil filter adaptor (see instructions above).

2. Remove nuts and washers from voltage regulator mounting bolts and retain for reassembly.

3. Insert cooler mounting bracket onto regulator mounting bolts. DO NOT PINCH ELECTRICAL WIRING. On some model(s), it may be necessary to unfasten the regulator wiring

from the retaining clip and to rotate the retaining clip and wiring in order to route the oil lines properly.

4. Reinstall the voltage regulator over the oil cooler mounting bracket and fasten using the nuts and washers removed in step 25. Use Loctite adhesive on all nuts and bolts.

5. Route the oil lines down from the oil filter adaptor installed in step 1.

6. Attach the oil line from the left port on the oil filter adaptor to the top port on the oil cooler. Attach the oil line from the right port on the oil filter adaptor to the bottom port on the oil cooler. Note: Oil cooler flow path is not directional. The suggestions for oil line routing are for ease of fitment.

7. Secure with the ⅞” black (color-coordinated), hose

clamps provided.

CAUTION: IF REGULATOR WIRING HAS BEEN DETACHED DURING INSTALLATION, INSURE REATTACHMENT OF REGULATOR WIRING.

8. Start the bike and check for leaks. 15

Conclusion

I hope I have rid the reader's minds of mythology surrounding the necessity of an oil cooler. Riders no longer need debate whether an oil cooler is necessary on air-cooled Big Twins and Sportsters for that matter. There is no reason whatsoever, to not run one or even two as I do. Moreover, I am not talking about those cute little chrome ones either, although they are better than nothing is. I want to obtain maximum heat exchange dissipation to eliminate any chance

of the great engine destroyer, detonation.

Oil cooling takes heat into safe territory.

1 Mike, my email buddy. Email. Detonation and Octane Nov. 29, 2004 5:31pm.

2 Ibid.

3 Wood, Bob. Wood Performance Carburetors. Part of a telephone conversation early 2007.

4 Beals, Marv. Email. Subject: Reply to questions.doc. Tuesday, January 08, 2008 11:21 am.

5 Beals, Marv. Email. Subject: Reply to questions.doc.

Tuesday, January 08, 2008 11:21 am.

6 Setrab USA, Inc.. Jagg Philosophy Creates Real Performance Advantage.

7 Ibid.

8 Beals, Marv. Email. Subject: Reply to Tuesday, January 08, 2008 11:21 am.

9 Website. (www.jagg.com). Compilation of information from various parts of the site including FAQ; Universal Oil Coolers; By-pass Valve and Jagg Designed Oil Cooler Products for Harley-Davidson Motorcycles.

10 Beals, Marv. Email. Subject: Reply to Two Pass Design. Tuesday, January 15, 2008 10:58 am.

11 Beals, Marv. Email. Subject: RE: Jagg Oil Wednesday, July 04, 2007 4:10

12 Beals, Marv. Jagg Philosophy Creates Real Performance Advantage. Email. Wednesday, September 05, 2007 4:46 pm.

13 Ibid.

14 Beals, Marv. Email. Subject: Reply to Tuesday, January 08, 2008 11:21 am.

15 Website. (www.jagg.com). Harley-Davidson. Adaptor 10 Row Vertical Oil Coolers, Installation. Deluxe Oil Cooler, Installation Slimline Oil Cooler.

# Chapter VII

## Torque and Horsepower

In general, torque expresses as the amount of work an engine can produce, which does not tell us much on a practical level in performancing a Harley-Davidson Twin Cam.

### Torque

Torque is the force that moves our Harley's is more defining and is self-explanatory. Better still torque is the force that accelerates the bike or produces acceleration overcoming a physics principle, the inertia of standing still.

The next definition explains how torque develops. Torque is the equivalent of the force created when the fuel mixture ignites and burns thereby creating expanding heat.

Heat is the same as pressure. Heat expands creating pressure. The expanding pressure forcing the piston down turns the crankshaft. Torque is the twisting force of the crank turning.

This torque creation transfers through the powertrain to the rubber patch contacting the asphalt on the rear wheel. The greater the torque and the lower in the rpm range it develops determine how fast the bike will accelerate off the line.

Torque measures in foot-pounds. One ft-lb of torque factually defines as one pound of force applied at a one-foot

radius. There will be more on this later.

## Torque and Horsepower Relationship

Torque is the twisting force that gets the motorcycle moving. Therefore, torque is all about acceleration.

Horsepower is the effort required to keep the bike moving. Therefore, horsepower determines speed.

I caution the reader that these concepts as described are overly simplistic much the same as viscosity of oil is the weight or thickness of the fluid. It really is not. However, most everyone thinks it is.

Torque really is not acceleration and horsepower is not

really speed but so many people think it is that I might as well acquiesce and not split hairs.

Torque can increase, stay the same, or decrease by changing one or both factors that determine the amount of torque.

Torque is force that applies to a lever.

Torque-Lever Examples

Suppose I take, a three-foot long bar and attach it to a socket on say, a compensating sprocket nut for illustration of this concept. I apply 50-lbs of turning force to the end of the bar. This will equal 150 ft-lbs (3-feet x 50 lbs), of turning force known as torque to the compensator nut. If I took a two-foot long bar attaching to the socket and nut, it would take 75

lbs of force to the end of the bar to produce the same 150 ft-lbs of torque (2-feet x 75 lbs), to the compensator nut. A one-foot long bar and socket will require 150 lbs force to the end of the bar to achieve 150 ft-lbs (1-foot x 150 lbs), of torque. If I still wanted to produce 150 ft-lbs of torque but only apply one-pound of force to the end of the handle, I would need a bar 150-feet long (150 feet x 1 lb).

Therefore, the force applied to an engine shaft by the rotating flywheels uses the same principle, as torque is a turning or twisting force. This torque measures easily with a dynamometer, colloquially known as a dyno.

Horsepower calculates without difficulty by dividing the engine rpm by 5252 and multiplying the result by the torque.

Horsepower

Torque is a specific measurement that easy to understand. Torque is precise.

A horse power is a very arbitrary measurement. Horsepower only seems easy and on the surface, it is. The term horse power can mean many things and measures differently in so many ways around the world, which I will go into later. It is always necessary to know which definition of horsepower that is in use.

It is amazing that this measurement of power survives today. The guy that invented a measurement for energy including light bulb wattage also defined horsepower. His name was James Watt (1736-1819). James Watt wanted to figure out how much work a horse could do removing coal from a perpendicular coalmine shaft. He experimented with smaller-than-normal coalmine horses. An average coalmine horse could do about 22,000 ft-lbs of work per minute.

In the USA torque measures in foot-pounds (ft-lbs) but in most of the rest of the world, it measures in Newton meters (Nm), sometimes written as Newton-meters or (N-m).

Watt then arbitrarily decided that a regular horse could do 50% more work in a minute than a smaller coalmine horse that he based his calculations on. Therefore, the figure became 33,000 ft-lbs of work per minute (22,000 + 50% = 33,000), or one horsepower.

Like torque, horsepower has two components that can increase, stay the same or decrease by changing one or both factors that determine the number of horsepower.

Torque factors are force and the length of the lever that the force applies.

Horsepower factors are weight and the distance it moves in one

Horsepower Examples

Therefore, a horse can move 330-pounds of coal a distance of 100 feet (330 x 100 = 33,000 ft-lbs) in one-minute producing one horsepower. Alternatively, a horse can move 100-pounds of coal 330-feet (33,000 ÷ 100 = 330-feet), in one minute producing one horsepower. On the other hand, a horse can pull for 200-feet a weight of 165 pounds (33,000 ÷ 200 = 165 pounds) in one-minute producing one horsepower. It matters not the distance or weight as long as the multiple of both arrives at 33,000 ft-lbs per minute to produce one horsepower.

It might seem that the following has not much to do with anything because it will seem absurd but it has practical applications in an engine demonstrating we stay within reasonable parameters.

## Horsepower Gearing

Could a horse move 33,000 lbs a distance of one foot in one minute to produce one horsepower? I think not. Conversely, can a horse move one pound of coal 33,000 feet in one minute? Well, I suppose the horse could, if it ran at 375 mph (33,000 feet x 60 minutes ÷ 5280 feet in a mile = 375 mph). Watt kept the mythical bigger horse and the real life coalmine horse's ability to do work within a reasonable range via pulleys to allow a consistent and even workload.

The performance mechanic makes the horsepower realistic and achievable through gearing whether it is taller for higher

rpm or lower for more twisting torque. However as every mechanic knows, there must be a happy medium or compromise in gearing to give both torque and rpm capability.

If an engine is all torque with extremely low gearing, rpm will certainly suffer. Therefore, it is easy to gain torque if rpm is unimportant.

Conversely, if an engine is all rpm the low-end torque will similarly suffer. Therefore, it is easy to gain rpm or speed if torque is not an important consideration

Thus, torque and rpm are interrelated.

A Harley-Davidson Twin Cam, particularly those 2006 and earlier have grunt off the line torque. As the reader surely

knows, the 2006 Dyna and all 2007 and later Twin Cams increase the gearing (taller), substantially.

A rider can start from a stop in gear and the Hog will pull forward, not that we would ever lug an engine in this way. It is important for a discussion that the reader compares this off-the-line power with a traditional high-revving Japanese bike, which can make much power in higher rpm levels but could not accelerate off the line in a higher gear.

Torque does not deal with how fast a Harley is moving but it makes the beast move under almost any circumstance. Torque is only a determination of how much twisting or turning force the rotating crank applies.

## Calculating Torque and Horsepower

Horsepower is the ability to reach higher speeds and hold

it there. However, there is torque at all speeds. Both torque and horsepower intertwine in reality and it easily demonstrates mathematically. Where there is one, the other will always be there. One cannot exist without the other. Torque and horsepower are always equal at 5252 rpm.

Circular Mathematical Constant: Pi

To understand the formula that derives at 5252 we need to understand the mathematical constant pi. One ft-lb of torque factually defines as one pound of force applied at a one-foot radius. Therefore, torque is twisting force at a radius, which is half the diameter of a circle. Thus, we need to know a little bit about a circle for the purpose of calculating torque and the intertwining concept of horsepower.

We know the Mathematical Circular Constant as $\pi$ (Pi).

Please be patient, this will not be that bad or difficult. The symbol for pi or Pi is π pronounced as pie and written as Pi or π. Pi is one of the most important mathematical constants and the magic number it represents is 3.14159. This figure is never ending, as it has no finalization. The number just keeps going. It never repeats. Therefore, π cannot express as a fraction.

This differs from many fractions such as 2/3, which repeats forever as .66666 and on it goes.

In Euclidean geometry, π signifies the ratio of a circle's circumference to its diameter. The Greeks sure figured out a lot of complicated stuff way back when.

Formula #1 for π

Pi equals the circumference of the circle divided by its

diameter

C = the circumference of a circle; D = the diameter of the circle

$$\pi = C \div D$$

This ratio of circle circumference to circle diameter always remains constant.

All circles have an enduring similarity that flows from the relationship of the circumference and the diameter or simply put the fraction of C over D (C/D). If a circle's circumference is half the size of another circle, the diameter will also be half. If a circle has three times the diameter of another circle, the circumference will also be three times greater.

Pi expresses this relationship as a constant mathematical figure.

We can also state this relationship as the ratio of a circle's area to the square of its radius A radius is half of a circle's diameter.

Formula #2 for π

A = area of the circle; r = radius of the circle

$$\pi = A \div$$

Pi is necessary in many formulae in mathematics, engineering, and science equations. Pi is very special. It is an irrational number and has other qualities that I will not go into here like in algebraic operations like roots, powers or sums that are incapable of producing π.

Pi comes from the Greek word which is logical since it deals with mathematical relationships of a circle to its diameter or a ratio of a circle's area to its squared radius.

The reader may see π referred to as Ludolph's Number, Archimedes Constant or simply as the Circular Constant.

How is the Constant 5252 Determined?

The formula to determine the constant 5252 is 33000 ft-lbs (one-horsepower) divided by 2 and then multiplied by pi.

$$1\ HP \div 2 \times \pi = 5252$$

$$33{,}000 \div 2 \times 3.14159 = 5252$$

The constant 5252 allow us to determine the horsepower of an engine.

Horsepower is literally torque times rpm.

We still divide the above figure by 5252 by convention, through tradition but it is unnecessary. If not using the scaling-down figure of 5252, we would only have to acclimatize to higher horsepower figures. It is relative; like which engine has more.

Remember that horsepower derives from the work a horse is capable of doing in a minute. The 5252 figure scales down the work one horse can do in one minute. No one ever discusses mechanics or an engine's capability in terms of one measly horsepower. However, all of my conjecture is just as irrelevant as to how the term and quantitative definition of

horsepower arbitrarily evolved.

The bottom line is that this is how engine power measures from day one and will continue to measure.

Interpreting the 5252 rpm Meeting Point of Torque and Horsepower

On Dyno graphs recording horsepower and torque, the lines recording both always cross at 5252 rpm. As I have said, torque and horsepower are always equal at 5252 rpm.

Above 5252 rpm, the torque will always be less than the horsepower.

Below 5252 rpm, the torque will always be more than the horsepower.

Cylinder fill and torque also intertwine inexorably. Cylinder fill is the percentage fill of the engine's cylinders with fuel comprising of air and gas. The fancy name for this is volumetric efficiency (VE). Maximum torque will always be at maximum cylinder fill (VE). This is common sense when one-steps away from all formula stuff I am about to explain.

Since torque is a measurement of the pressure forcing the piston down thereby twisting or turning the crank, it only makes sense that the most cylinder fill of air and gas will make the most torque. Therefore, it is also logical that torque will fall off when volumetric efficiency or cylinder fill lessens.

As long as the flow capability of the carburetor or EFI as the case may be, the manifold, the head ports and the valves will allow more air we can increase both the torque and

horsepower. A subject that I will return to repeatedly is that one might logically think that the optimal volumetric efficiency of a cylinder would be 100%. Through headwork and longer duration cams that opens the valves for a longer time beyond the theoretical engine rotation, combining with high rpm, the principles of momentum or scavenging will come into play. Exiting exhaust gases will act as if there is a vacuum behind them and help suck in fresh intake gases at a greater rate than normal. This is sort of like a reverse supercharger. The fuel charge packs into the cylinder beyond 100% fill.

It is not out of the question to obtain up to 130% fill.

Scavenging does not occur at lower rpms because the exhaust gases are not moving fast enough to act like a vacuum. Therefore, the maximum torque moves up the rpm

range because the maximum cylinder fill has moved from lower rpm to higher rpm.

Since all things are not equal, the modifiers to this general discussion other than the intake components listed above will be combustion chamber configuration, squish bands, spark plug location, number of spark plugs per cylinder, compression, engine head temperature and quality octane gas.

Horsepower and torque have the mathematical relationship that determines if torque increases then so will the other at a specific constant rpm. However if torque remains constant over an rpm range then horsepower will either increase or decrease in direct proportion to the rpm change. Increasing rpm will develop more horsepower if the torque curve remains constant. Lowering rpm will decrease horsepower with constant torque. If rpm stays the same then the only way to increase horsepower is to increase torque.

I realize I am going around this subject and viewing it from different angles but it helps in the understanding of this most misunderstood subject matter. One last kick at the can will suffice.

Producing horsepower at a lower rpm requires more torque than it takes at higher rpm.

Since torque is, a measurement of the pressure generated by the expanding heat of burning fuel after ignition it makes sense that more cubic inches will produce more torque. More cubic inches allows for more cylinder fill.

On the Evo engine, VE can increase by porting and polishing the existing heads or installing performance heads. This will also allow more cylinder fill or volumetric efficiency.

However, the Twin Cam heads are another ballgame as they are inadequate for the size engine so this becomes a second way other than cubic inches to raise cylinder fill.

Formulae for Horsepower and Torque

The first formula will allow the calculation of horsepower when we know the torque and rpm.

$$HP = Torque \times rpm \div 5252$$

Example # 1; a Harley-Davidson TC88 produces 85 ft-lbs of torque at 3000 rpm.

$$HP = 85 \times 3000 \div 5252$$

HP = 48.6

Example # 2; a Harley-Davidson produces 95 ft-lbs of torque at 3500 rpm.

HP = 95 x 3500 ÷ 5252

HP = 63.3

The second formula will allow the calculation of torque when we know the horsepower and rpm.

Torque = HP x 5252 ÷ rpm

Example # 3; a Harley-Davidson TC96 produces 52.6 HP at 3000 rpm.

Torque = 52.6 x 5252 ÷ 3000

Torque = 92 ft-lbs

Example # 4; a Harley-Davidson TC110 produces 86 HP at 4000 rpm

Torque = 86 x 5252 ÷ 4000

Torque = 113 ft-lbs

The Interrelationship between Torque, RPM, and Gearing

It is important to understand that the basic relationship between torque and rpm is that with lower gearing torque will increase. Torque will decrease with higher gearing. In fact,

the relationship is equal in that if one goes down the other will go up equally and vice versa.

Torque and rpm are interchangeable!

Now, I will state this in the opposite way, which will help us even more in performancing not only the Twin Cam but also any engine.

Therefore, gearing becomes the key in adjusting torque figures up or down. It follows that gearing is also the solution in increasing or decreasing the rpm capabilities of a Harley-Davidson.

Why the TC96 Horsepower Increase is Disappointing over the TC88

Ergo, the TC96 revs higher than a TC88 because it has

higher gearing. We now have the explanation for disappointing horsepower results from the eight cubic inch increase in the TC96 over the TC88. The TC96 produces 52.6 horsepower at 3000 rpm while the smaller TC88 produces 48.6 horsepower at the same 3000 rpm.

The gain is a paltry four horsepower with eight cubic inches.

However, the TC96 is very much faster than the TC88 particularly at higher rpms. Furthermore, torque is not a meaningful concept unless including the rpm.

Now, we can also understand that there is more torque and lower rpm in gear over gear at the rear wheel, which has more torque and less rpm than gear, and on it goes until we get to of gear depending on the stock transmission, which has

a 1:1 final ratio. It is easy to pull a wheelie in gear because of lots of torque with lower rpm capability and not much prospect of exceeding 30 or 40 mph. Up shifting, the transmission reduces some of rear wheel torque but increases rear wheel rpm. Conversely, it is impossible to pull a wheelie in where rpm reins and torque is minimal but the TC96 is capable of maybe 115 mph, which the stock TC88 cannot match in its high gear of fifth.

Performance Requires Increases to both Torque and RPM

It is impossible to increase torque without lowering rpm when downshifting the transmission and not possible to increase rpm without lowering torque while up shifting.

The trick is to increase both at any given speed.

Horsepower is the term given to increasing torque and rpm

at a given speed.

Do I sound smart? Not really, just look at the formula above for horsepower; it is rpm x torque divided by a constant that does not affect the relationship between torque and rpm.

If the increase in torque and rpm is not in a desirable area, then gearing will adjust it up or down.

## Compression

Compression ratio is a catchall phrase. There are a whole bunch of ways of looking at compression measurements and even more terminology referring to all this stuff.

I need to review a few like corrected compression ratio. In

order to understand what corrected compression ratio refers to; I will look at static compression ratio first.

## Static, Theoretical, Mechanical, Advertised, or Nominal Compression Ratio

The static compression ratio also known as theoretical or mechanical compression ratio as well as advertised or nominal compression ratios compares the volume of the cylinder and combustion chamber to the volume of the combustion chamber alone.

### Calculating Static Compression Ratio

Example: 865cc cylinder and an 89cc combustion chamber.

Static Compression Ratio = Cylinder + Combustion chamber volumes

Combustion chamber volume

Static Compression Ratio = 865 + 89

89

Static Compression Ratio = 10.7:1

To calculate the static compression ratio I add 890 + 95 = 985. I then divide 985 by 95 = 10.4:1 to obtain the static compression ratio.

Corrected or Effective Compression Ratio

The corrected or effective compression ratio is more

accurate for a real-life engine. It recognizes that cylinder volume is lost with open intake and exhaust valves during the initial part of the compression stroke. The math to calculate this is complicated requiring a computer program.

## Cranking Compression: Dry and Wet

Then we have cranking which measures in psi (pounds per square inch), actual compression with a tool called a compression gauge. The technician screws in or presses a hose connecting to a gauge into the spark plug hole and cranks the engine over to get a dry cranking compression in a psi reading.

Wet cranking compression is usually higher than dry testing because the technician squirts oil into the cylinder beforehand so the rings can seal better.

## Running Compression

Running compression is what it says. Running compression is actual real-time compression under riding conditions where there is heat expansion and ring sealing against cylinder walls with oil lubrication.

## Altitude and its Effects on Compression

Altitude lowers corrected compression ratios because there is progressively less air with rising elevation. Altitude will not affect static compression ratios, as it is just a comparison of volumes in the cylinder and combustion chamber relative to each other.

To counter the loss of effective compression from longer cam durations and overlap, thinking mechanics will increase static compression ratios.

Torque is synonymous with engine compression, more specifically cranking

Compression Equals Torque

Compression also Equals Cylinder Fill

Compression depends on cylinder fill. Cylinder fill of 100% is optimal at lower rpms but can easily increase to 130% by stacking in the air and fuel via the scavenging forces of momentum as exiting exhaust gases help suck in the incoming intake charge while both valves are open during the overlap period.

With increasing cylinder fill comes increasing cranking compression since the piston must compress a bigger volume of air and gas molecules. Of course, the burning of this increase in cylinder fill and thus cranking compression creates more heat, which expands crating more pressure to drive the pistons down on their power strokes, rotating the flywheels with more twisting force known as torque.

Cylinder Fill Equals Torque

The reason torque drops off with increasing rpm is that the

piston must make its journey many more times per minute. The increasing piston speed makes complete cylinder fill more difficult, which reduces compression thus reducing torque.

Now there is a need to position the power band. We want to maximize cylinder fill with the chosen rpm range of the power band.

Maximizing cylinder fill depends on many support components in the engine.

I would venture an educated guess that most Harley riders will want to increase cylinder fill at lower rpms to produce horsepower comprising of more torque and lower rpm.

Lowering the Power Band

If I look at say, Bobby Wood's cam designs (www.woodcarbs.com), they are renowned for lowering the power band into very useful rpm ranges. His grinds use shorter duration profiles, which determine the time the valves are open than most other manufacturers and increase the lift, which is how much the valves open relative to the duration.

Shortening duration is overly simplistic as Bobby also depends on overlap valve timing, positioning of cam lobe centerlines and thus the lobe separation angle to develop low rpm torque.

Many of today's aftermarket cams including Screamin' Eagle (www.harley-davidson.com), generally have longer duration profiles, differing overlap valve timing and wider lobe centerline positioning and thus a wider lobe separation

angle relative to lift that take away from low bottom-end grunt torque for off the line power. These longer duration cams work wonderfully in the higher rpm stratosphere riding that can challenge the mechanical limitations of pushrod-operated engines. Do not get me wrong duration is a good thing. It is a strong component in the equation for more power derived from camshaft lobe design.

However, more is not always better. Too little duration as exemplified by EPA dictated stock cams is also not conducive to maximum power either. These EPA cam profiles are all about containing emissions. Power is a secondary consideration.

How to maximize torque

It is pretty simple; we need maximum cylinder pressure for maximum torque. Torque multiplied by rpm gives

horsepower.

## Horsepower is a Multiple of Torque and rpm

Always remember that horsepower consists of a multiple of torque and rpm. Therefore, more torque with less rpm will multiply to the same horsepower as an equivalent higher rpm and less torque. For example, 80 ft-lbs of torque at 4000 rpm equal 60.9 horsepower, (80 x 4000 ÷ 5252 = 60.9).

However, at 2000 rpm, an engine producing 60.9 horsepower will have 159.9 ft-lbs of torque, (Torque = 60.9 HP x 5252 ÷ 2000 rpm = 159.9 ft-lbs)

Let us confirm this figure by the first calculation to determine horsepower. Torque of 159.2 ft-lbs at 2000 rpm must equal 60.9 horsepower. Therefore, multiply 159.9 ft-lbs torque by 2000 rpm, which equals 319,800. Divide 319,800

by the constant 5252, which equals 60.9 horsepower.

Maximizing cylinder fill and thus cranking compression, which increases torque, will increase horsepower.

There is a variety of ways to do this. Coordinating the various components in the engine with each other in a supportive way to increase cylinder fill will exponentially increase cranking compression.

Head Ports and Valve Sizing

The factor of correct sizing and shaping of head ports relative to the size of valves coordinated with the swept volume capacity of the cylinder is necessary to increase cylinder fill and thus cranking compression and the resultant torque increase. This process increases flow of air and gas into

the cylinders.

The inherent performance obstruction in both the Twin Cam 88 and the Twin Cam 96 engines is lack of sufficient flow into the cylinders. Cylinder fill is not what it should be. Therefore, cranking compression suffers. Then torque does not maximize. Finally, the amount of horsepower suffers.

A shaping adjustment or modification of the head ports through the process of porting accomplishes by grinding but sometimes involving the addition of metal via welding. Porting is the reshaping of intake and exhaust tracts in the cylinder heads while polishing is a process usually only employed in the exhaust tract to facilitate the fast exit of exhaust gases. I am going to describe in detail porting and polishing necessities to increase flow in both the TC88 and TC96 head castings in Chapter VIII, Eliminating the

Performance Obstruction.

Mixture of the air and gas is also crucial. If too fat, where there are too many gas molecules relative to the oxygen in the air, the mixture will not ignite. If too lean where there are too many oxygen molecules relative to gas molecules, the mixture will have difficulty igniting. If it does burn, the heat will be searing.

Head port shape and texture has some influence on initial mixing of molecules although the bathtub-shape combustion chambers in all Twin Cam engines have 360-degree squish bands that the flat-topped piston squishes against to force the air and gas turbulently into the smaller combustion chamber giving an excellent mix of molecules that burns easily.

A Smaller Valve can Increase Velocity and Flow

Physics principles such as Bernoulli`s Principle come into play with port shape and valve sizing. Watch a wide lazy river as it narrows. The water becomes turbulent and moves faster. Conversely, if a rushing white water river gorge opens up into a wide plain, the water smoothes out and slows down. Bernoulli figured out that the weight of the water flow always pushing from behind would determine speed of a constant volume of flow. In other words, the same volume of water must move in the same time-period no matter the size of space through it must flow. The lazy wide river has much volume that must pass through a narrowing in the same time. The only way the water can accomplish this is to speed up. This is why it also slows down when approaching a widening from a narrows.

Velocity increases flow into the cylinders and thus cylinder fill, cranking compression, torque, and then horsepower.

The reader will never see a head port in a naturally aspirated engine that becomes wider as it progresses from the induction manifold to the valve. The velocity of gases will decrease. This is also, why a smaller valve may produce greater flow through velocity than a larger valve. Once again, the mechanic learns that bigger is not always better.

Naturally aspirated means that the natural laws of physics like vacuum and momentum determine engine breathing while forced aspiration such as supercharging or turbocharging are very effective but artificial means of induction.

The key, once again, is the correct sizing and shaping of head ports relative to the size of valves coordinated with the swept volume capacity of the cylinder, which is necessary to increase cylinder fill and thus cranking compression and the resultant torque increase that along with rpm determines horsepower.

# Compression Ratio vs. Cam Duration

A rule of thumb is that more cubic inches can absorb more duration. Cam duration is how long a valve stays open before and during its theoretical 180-degrees of crankshaft rotation. An intake valve that opens BTDC (before top dead center), before the intake stroke during the last part of the exhaust stroke and closes ABDC (after bottom dead center), in the initial stages of the compression stroke after the completion of the intake stroke has a duration of + +

The three main factors determining cylinder pressure are the compression ratio modified by camshaft duration and altitude. Higher altitudes lower atmospheric pressure because there is not as much air. Therefore, less air results in less compression. As a generalization, the higher the altitude the

less camshaft duration is necessary. This does not apply to visiting higher altitudes, as cam duration reduction is not possible without replacing the camshaft. Less duration is necessary when living and riding at higher altitudes most of the time. Increasing cylinder pressure results from more compression and less cam duration. Conversely, a decreased cylinder pressure is the result of less compression and more cam duration.

Therefore, in performance applications more cam duration is necessary to bring in more air to burn more fuel, which creates more pressure to drive the piston down on the power stroke with increased force to turn the rear wheel with the following caveat. More cam duration will necessitate a higher compression ratio to maintain a higher cylinder pressure to maintain an increase in power.

Longer duration cams keep the valves open longer and generally require an increase in static compression because cranking compression begins to build later as the piston rises in the cylinder on the compression stroke. Regaining this lost pressure due to longer valve openings can theoretically offset by increasing the compression ratio. However, all the compression lost through duration and regained by a compression ratio increase may not fully compensate and still cause a decrease in power.

## The Volume to Pressure Index (V/P Index)

The compression gauge may indicate equalization but low and mid power may still suffer. There is still a smaller volume of air/gas compressing to a higher ratio. The compression gauge accurately reads to an equal or even higher compression than before the installation of a longer duration cam and an increase in static compression but in fact the cylinder pressure multiplied by the actual captured air/gas

mixture may still be lower than the original mild cam and lower compression. In conclusion, it is very necessary to increase compression when installing a longer duration cam but it may not totally offset the effects of the valves being open longer in lower rpm levels. However, power will certainly increase at higher rpm levels via the effects of scavenging and increasing cylinder fill.

The volume to pressure index known as the V/P Index is a calculation that monitors this situation but cannot predict maximum torque or horsepower.

## Cam Timing Events and Cylinder Fill

Cam timing or when the valves open and close in the overlap period relative to each other have much effect on cylinder fill, cranking compression, torque and horsepower.

The term valve events or cam timing events describe when valves open and close. The cam timing events are the Intake Valve Opening position, Intake Valve Closing position, Exhaust Valve Opening position, Exhaust Valve Closing position, Valve Lift, Overlap and finally the Lobe Separation Angle.

The only cam-timing event affecting cylinder pressure directly is the Intake Valve Closing Position. This is the point where the intake valve has just closed after bottom dead center (ABDC), of the intake stroke at the beginning of the compression stroke. This intake valve closure during the beginning of the following compression stroke will always cause some of the intake mixture to compress or reverse (reversion), backwards out of the cylinder.

Peak Torque

The air/gas reversion unproductively escapes back through the still open intake valve into the intake tract driven by the rising piston at the beginning of the compression stroke. This occurs at low speeds as the piston is just beginning to rise on the compression stroke and is not yet at full velocity. Radical, late intake valve closure ABDC will logically increase this mixture reversion both at low-speed and into moderate speeds. This, of course, reduces cylinder pressure and thus compression at the bottom of the torque curve. The torque curve will reposition upwards.

Peak torque generally occurs at the engine rpm where this intake reversion stops and all the air/gas traps at stroke completion. However, as always, the above general statement mitigates by other factors such as the other cam timing events, head port efficiency relative to valve size and cylinder

volume and intake charge velocity to name a few.

A simple event viewed in isolation gets complex very quickly when entering the mix of other mitigating factors. Therefore, the greatest torque achieves by closing the intake valves early into the compression stroke and secondly by not opening the exhaust valves too early in the latter stages of the power stroke. We do not want potential compression leaking off because valves are open at the wrong time. Maximum cylinder pressure and thus torque occurs approximately between and after top dead center (ATDC) of the compression stroke.

## The Cam Lobe Centerline Angle (LCA) Effect on Cylinder Fill

Lobe Centerline Angles (LCA) determines two important valve-timing characteristics. The LCA or LC is the highest point of the cam lobe. Technically, there are four LCA's in a

Harley-Davidson because there are four cam lobes, each governing a valve. However, the intake and exhaust cam lobes in each cylinder are identical to each other in a Harley. Therefore, there is only two LCA's, an intake LCA and exhaust LCA.

The LCA dictates the valve overlap around top dead center (TDC). Secondly, LCA establishes the amount of intake or exhaust valve closure delay there is beyond the theoretical end of the applicable stroke.

Hi-torque cams have lobe centerlines around For example, my current Knight Prowler TW408G cams have an intake centerline and exhaust LC of

A tight LCA increases cylinder pressure and thus torque with a rougher idle and the danger of exhaust reversion back

into the chamber bastardizing the fresh intake charge.

A wider Lobe Centerline Angle causes the intake valve to stay open longer after bottom dead center (ABDC), of the intake stroke decreasing cylinder pressure and thus torque.

A stock engine will receive up to an approximate 10% fill after intake BDC while performanced engines receive more fill, possibly up to and past 20% extra fill.

Greater increases are negated by the now rising piston on its compression stroke.

Too wide an LCA produces the possibility of intake reversion where the intake air/fuel charge may force back up and out of the intake tract reducing potential power, specifically

As I said earlier and it bears repeating that, the Lobe Centerline Angle affects valve overlap around piston top dead center. In addition, the LCA establishes the amount of intake or exhaust valve closure delay there is beyond the theoretical end of the relevant stroke.

As the piston descends to the bottom of the cylinder but before the piston reaches the bottom, the exhaust valve begins to open. Most of the fuel has burned. Cylinder pressure will begin to push these burnt exhaust gases out into the exhaust port and the exhaust headers into the mufflers. After the piston passes bottom dead center (BDC), it begins ascending the cylinder on the exhaust stroke. This upwards action forces the remainder of the spent exhaust gases out of the cylinder and combustion chamber to make room for the fresh, clean

charge of air and gas. As the piston moves toward the top of the cylinder, the already opening exhaust valve goes through maximum lift and begins to close. Just before the piston reaches the top of its exhaust stroke (TDC), the intake valve begins to open but the exhaust valve is not yet fully closed.

Theoretically, this should not work unless other forces of physics assist like the Principle of The piston's upward exhaust stroke is removing just about all of the spent charge by the time the piston approaches top-dead-center. However, just before this, the intake valve begins to open slowly. Scavenging (Principle of Momentum) begins. The remaining exhaust-gases rushing out into the exhaust port and header will draw or suck in the start of the intake charge.

Conversely, the developing momentum of the intake charge also assists in pushing out the remaining exhaust gases.

It becomes a symbiotic affair, first with exhaust pulling and then a little intake pushing. Scavenging is how the engine flushes out all of the spent fuel charge.

Again, some of the new intake charge gases may escape into the exhaust. Thus, valve opening and closing timing is crucial. Finding the sweet spot where pollutants fully exit but the beginning of the fresh intake charge is not lost is the science and many times the art of valve timing.

Once the piston passes through the exhaust stroke ending theoretically at TDC and starts back down on the intake stroke, the intake charge pulls in quickly. Therefore, the exhaust valve must close at precisely the right point after the TDC to ensure prevention of any burnt gas re-entering.

Overlap is this area around TDC where both intake and exhaust valves are open. This is one of the most critical

moments in the engine's rotational cycle. Intake and exhaust valve opening and closing must position correctly with the top dead center of the piston.

To complicate matters the function design of the motorcycle changes this positioning for different types of riding, performancing modifications, and other factors. Different types of riding demands different overlap periods, as do other changes.

As the engine passes through the overlap phase, the intake valve is opening very quickly while the exhaust valve closes just after the piston starts descending. As the reader knows, the intake stroke is where the engine breathes in yet another fresh charge of virgin air/gas mixture.

The intake valve reaches its maximum lift at some planned

point, which is in and around 105-degrees of crankshaft rotation after top dead

As I discussed earlier, this degree-measurement is the intake centerline lobe also known as the lobe centerline (LC) or Lobe Centerline Angle (LCA). This refers to the position of cam installation also known as degreeing in the engine in relation to crankshaft rotation.

All moving engine components time off the positioning of the crankshaft.

LCA determines valve overlap around piston TDC and has much influence on the amount of intake or exhaust valve closure delay there is beyond the theoretical end of the relevant stroke. Furthermore, tighter or closer LCAs produce earlier intake valve closure after BDC. A hi-torque low rpm cam needs tighter LCA's. Wider or further apart Lobe

Centerline Angles extend the intake valve closing beyond BDC. This will move the peak torque of the power band up the rpm range.

## Cylinder Pressure and Intake Valve Closing

The piston descends all the way to BDC.

As the piston begins its ascension in the cylinder bore on its compression stroke, the intake valve is fast returning towards its mating valve seat. The closing point of the intake valve will determine where the cylinder actually begins to build pressure.

After the intake, charge is in the cylinder, the intake and exhaust valves close. However, it is very important to have the intake valve open long enough to fill the chamber. It is

equally important that it close it soon enough to yield maximum cylinder pressure.

This is a very tricky valve-timing point in the camshaft cycle.

Corrected cylinder volume is the cylinder displacement trapped when the intake valve closes.

The power builds as the ascending piston begins to compress the mixture. Compression, where the piston begins to build pressure is crucial to power.

Compromise Spark Ignition Timing

Before TDC on the compression stroke, ignition sparks the plug.

Ignition timing is how far before TDC ignition occurs on the compression stroke before the power stroke begins. Ignition timing will change frequently as it follows its ignition curve. This advance or retard ignition timing is necessary because of many factors needing more or less time for the flame travel to travel across the combustion chamber.

Accelerating the bike means the piston is moving faster so spark must occur sooner also. Conversely, decelerating the bike allows the spark to ignite later.

A big bore kit whether it is the TC88 to TC95 or the TC96 to TC103 conversions will increase the diameter of the cylinder to create more swept volume or displacement. The flame travel needs more time to travel the extra distance so the spark must ignite the fuel mixture sooner.

These few examples are some of many. All the various factors affecting ignition timing will end in compromise spark timing.

Decelerating rpms and thus later spark ignition must compromise with the earlier spark timing necessary for big bore and on it goes through a myriad of demands.

In conclusion, from a broad perspective, lobe centerline angles strongly tie in with the flow capabilities of the cylinder heads relative to the cylinder swept volume needs. Many refer to this as the ratio of cubic-feet-per-minute (cfm), air per cubic inch displacement, (cfm per c.i).

The timing advance gives both higher cranking compression and running compression resulting in more torque and horsepower in the lower rpm range.

Furthermore, it prevents chain driven cams from lost cam motion, or cam wander in the loose roller chains resulting in cam timing retard.

Compression ratios affect cam choice, especially with duration specifications, more than the size of the engine although displacement is also a factor. Higher compression and long durations work well together. Longer duration cams do not do well with lower compression. This is why many cam manufacturers' will specify compression ratio ranges suitable for their longer duration cam profiles. Conversely, longer duration cams will allow easier starting of high compression engines because these cams hold the valves open longer allowing some volume in the cylinders to escape.

Thus, the real life corrected compression ratios are lower

than the static compression ratios, which only compare the volumes of the cylinder and combustion chamber.

## Lobe Separation Angle (LSA)

LSA is short form for Lobe Separation Angle also known as the Lobe Displacement Angle or the Lobe Spread. A tight LSA has the respective intake and exhaust cam lobes (centerlines), closer together. A wide LSA has a wider angle between the high points (LC's), of the respective cam lobes. Other factors equal, tightening the LSA produces more valve overlap around TDC, while widening the LSA results in less overlap. At Bottom Dead Center of the intake cycle, a wide LSA produces a longer delay to valve closure after the piston has passed BDC

The calculation of LSA accomplishes by adding the intake and exhaust Lobe Centerline figures and then dividing their sum by two.

Overlap

Overlap directly controls engine cylinder vacuum, which is a necessary function for cylinder fill.

Do not confuse Lobe Separation Angles (LSA) with overlap.

In review, ably assisting engine vacuum that initially brings in the intake charge is the scavenging effect, a colloquial mechanical term, more properly known as the physics Principle of Momentum. As I say repeatedly,

scavenging creates the high velocity momentum of exhaust gases rushing out the exhaust port to create more vacuum suction behind the exiting exhaust gases that pulls the fresh air/gas charge into the chamber at higher rpms. This is very important since scavenging directly relates to the cylinder heads, exhaust system, and engine breathing capabilities.

If there are weak cylinder heads in terms of flow like those found in the TC88 and to a lesser extent in the TC96 and an improper poor scavenging exhaust system, the most obvious example being drag pipes on a street bike the wrong LSA might not put enough fresh charge into the cylinders.

Typically, a wider Lobe Separation Angle of to 116-degrees separation between lobe centerlines will mean less overlap and higher vacuum at idle. A narrower lobe separation angle to provides just the opposite. With street cams, the wider the

lobe centers move apart, increasing the Lobe Separation Angle such as to 116-degrees, the less overlap typically occurs.

The overlap phase begins with the opening intake valve. The timing of this is very critical to vacuum, throttle response, emissions, and particularly, gas mileage.

One of the most significant markers in the engine cycle is the area between the intake opening and the exhaust closing is the amount of overlap. Where it occurs is also crucial. If the intake valve opens, too early, exiting exhaust gases will push the new intake air/gas charge into the intake manifold. If intake valve opens too late, the cylinder will lean thus negatively hampering engine performance. If the exhaust valve closes too early it will trap some of the spent gases in the combustion chamber bastardizing the new intake charge. If the exhaust valve closes too late it will over-scavenge the chamber; taking out some of the fresh intake charge, again creating an artificially lean condition.

If the overlap phase occurs too early, an overly rich condition will develop in the exhaust port, negatively affecting gas mileage, which will be a big no-no to emissions controls. Therefore, it becomes easy to understand that everything about overlap is critical to the performance of the engine.

Hi-torque cams never have more than a separation between the exhaust Cam Lobe Centerline and the intake Cam Lobe Centerline.

## The Exhaust System, Cylinder Fill, and Torque

A narrow lobe separation angle will help increase cylinder fill, which increases cranking compression, which will deliver more torque.

Correct exhaust design for a specific engine will assist in all of this by ensuring exhaust gases leave the engine in a timely fashion and will prevent spent gases from polluting the fresh intake charge. The exhaust must be a suitable one or it will work against the engine and any potential power increases. There is no one-exhaust system that is superior neither in every situation nor for every Harley-Davidson engine especially when performancing enters the equation.

The stock exhaust certainly is not appropriate because its primary concern is to reduce emissions not allow an engine to breath freely, thus promoting efficiency and synonymous power.

I will radically demonstrate this with drag pipes. Drag pipes will decrease power in the lower and mid rpm ranges on any street Harley-Davidson for reasons that will become

evident later in this section. However, the same pipes will assist greatly in ridding a top fuel H-D dragster of unwanted exhaust gases thus creating power.

Different size engines need different diameter and length exhausts. The muffler will also need to increase in volume capacity as a street engine gains cubic inches or swept volume. Generally, a muffler should have ten times the volume of the cylinder it serves.

Complicating the exhaust valve closing timing by a narrow cam-lobe-separation-angle are kinetic wave pulses of exhaust gases returning up the pipes.

Kinetic Exhaust Gas Pulse Wave Theory

I am not sure why Kinetic Pulse Wave Theory has the word

theory as part of its nomenclature because what I am about to describe happens thousands of times every minute in your Harley and every other internal combustion engine.

Many think that the exiting exhaust gases simply flow out the exhaust header into the muffler and out into the atmosphere. Well they do, that is after they turn around and return up the pipe many times. This sounds dumb. Why would they do that?

One reason is that heat or temperature differentials create negative pressures (vacuum) that are greatest near the exhaust manifold with positive pressure near the muffler. This is just a moderately fancy way of saying that it is hotter at the exhaust manifold and cooler near the end of the muffler. Heat expands and cold contracts. The exiting exhaust gases cool as they go out the header and muffler. This creates a negative pressure back near the exhaust port causing the cooler positive exhaust to turn around in the muffler and travel back up the exhaust

reversing the flow of that particular exhaust pulse.

If depicting the traveling exhaust pulses on a chart or graph, they would not appear as a straight line but as a series of waves. Kinetic Theory is complex to the uninitiated but the bottom line for our purposes is that returning exhaust pulses must bounce of a closed exhaust valve to prevent polluting the fresh intake charge in the combustion chamber and cylinder.

Exhaust systems are perhaps the most complicated design part of a motorcycle or car. I watch riders in disbelief at their ignorance as they pull out baffles, install baffles, drill holes through baffles, remove exhaust plugs, cut pipes shorter, make them longer, increase their diameters and on it goes.

The problem is that they do not have a clue what they are doing. I am only knowledgeable enough to realize I do not

have sufficient knowledge in this very sophisticated arena. In short, exhaust design is way beyond the realm of technicians, mechanics, and most engineers.

All of these changes will change the kinetic exhaust pulse waves return timing back up the pipe. I hope that the pulses will bounce off a closed exhaust valve but perhaps the novice fooling with his or her exhaust will create the problem of exhaust pulses entering back into the combustion chamber past an open exhaust valve.

Tight Lobe Centerlines resulting in a narrow Lobe Separation Angle increases cylinder pressure and thus torque. The negative consequences of this are a rougher idle and the danger of exhaust reversion back into the chamber bastardizing the fresh intake charge. Tight LC's and the consequent narrow LSA will open the exhaust valve later and

will close it later. The later closing creates the danger of exhaust reversion because in this scenario the intake valve also opens earlier.

This creates greater valve overlap.

If exhaust reversion becomes an issue, then changing the degreeing of the cam lobe centerline may be necessary as a slight adjustment to balance out potential exhaust reversion. Once again cam design becomes art, where finding the fine line between increasing torque balances between the potential problems of exhaust reversion.

It is funny how some things will tie together.

I am about to discuss torque valves also known as anti-reversion (AR) valves that fit into or are designed into the header pipe of an exhaust system. The names never really

made sense to me before. Torque valves prevent reversion of exhaust gases back into the combustion chamber as they travel back up the pipes. A torque valve looks like an inverted cone where the wide diameter end meets the exhaust port. The narrowed end faces into the exhaust pipe header catching many of the pulse waves coming back up the pipe and rebuffing them, many times breaking the wave pattern.

Therefore, a torque valve preserves engine torque by preventing exhaust gas pulse reversion back into the combustion chamber.

I figure out much as I write for my own knowledge base.

Torque valves sometimes cast into the heads exhaust ports where they meet the exhaust header pipe. The Twin Cam TC88 head is a case in point. The exhaust port end is not

round but squishes in appearance. This design rebuffs returning pulses, again breaking up or disrupting the wave pattern.

The TC96 head eliminates this feature, so the exhaust pipe will have a design feature to assist in this crucial prevention of exhaust reversion thus preserving engine torque.

## Connecting Rod Ratios

What in the world, does a connecting rod have to do with horsepower or torque?

Furthermore, what is a connecting rod ratio? There is no ratio. The con rod is a fixed length.

Most mechanics do not fully appreciate the connecting rod ratio factor if they understand it at all.

The crankpin is offset from the sprocket and pinion shaft center axis by exactly one-half the stroke length. Sounds complicated, I know, but half the stroke creates at the top of the stroke or piston travel and the other half at the bottom of the stroke at the bottom of piston travel. Therefore the crankpin is only one-half the stroke length away from the center pinion and sprocket shafts because as the flywheels rotate the crankpin is half way above at say, the 12-o'clock position and half way below at the 6-o-clock position (½ + ½ = 1).

There is a geometric relationship between the connecting rod length and the crankpin stroke. Many times, most mechanics and riders ignore it because they just do not understand the correlation even if they have ever even heard of it.

However, the ratio between the two plays a key role in the engine's breathing capability. Breathing capability of the engine is all about power and power potential.

Connecting Rod Ratio Formula

$$\text{Con Rod Ratio} = \text{Con rod length} \div \text{Engine stroke}$$

Example, TC88 con rod length is 7.762" and the stroke is 4.000"

$$\text{TC88 con rod ratio} = 7.762 \div 4.000 = 1.94{:}1$$

The ratio of rod length to stroke length is generally between 2.2:1 on the high end and 1.4:1 on the low end.

Shorter connecting rods relative to the stroke produce low connecting rod ratios of 1.40:1 to 1.75:1.

Longer connecting rods in relation to the stroke generate high connecting rod ratios of 1.75:1 to 2.20:1.

From the accompanying chart, it is evident that most Harley-Davidson's, whether they be Flathead, Sportster or Big Twin including the Twin Cams range between a low of 1.75:1 for the late 1340cc Shovels and the 1984-1999 Evo Big Twins and the high end of 1.95:1 for the 45 Flathead and Ironhead Sportsters. The Twin Cams are on the low end of other Harley-Davidson's at 1.77:1 but are really in the middle of accepted engine con-rod-ratios.

Already, we should see that perhaps lower con rod ratios improve engine efficiency.

Flatheads and Ironhead Sportsters, as great and cool as they can be, are not known for reliability or engine efficiency whereas the late Shovels improve over earlier ones while Evos and Twin Cams dramatically improve.

Con rods measure center-to-center of the big end hole and the small end wrist pin hole. It is not possible for the connecting rods to equal the length as the stroke. Connecting rods are not practical if much more than twice the length of the stroke. The engine will literally become too tall for most practical street usage.

Con Rod Ratio Limitations

It is not possible for the connecting rod to be the same length as the stroke, which theoretically yields an unfeasible con rod ratio of 1:1. Probably, the only time where connecting rod length is more than twice the stroke is in a racing

application. However, as I said, the engine becomes very tall with a con rod ratio greater than 2:1. This ratio or greater is not practical for most purposes.

## Connecting Rod Length Affects the Engine's Power Band

Connecting rod length has a huge effect on the power band. The longer the con rod is the higher the power band will position in the rpm scale. I compare the effect of a longer connecting rod to increasing a cam's duration on the power band. Shorter connecting rods have the same effect as a shorter duration cam in lowering the power band. Therefore, shorter connecting rods will have a big effect on increasing bottom end torque.

Longer connecting rods are best for higher rpm motorcycles. Obviously, increasing the rod length will

increase the ratio but so will shortening the engine stroke.

Conversely, decreasing the rod length will decrease the ratio but so will lengthening the engine stroke. Wow, I was compiling the following chart when the best example of this just jumps out! The TC96, TC103 and CVO 110 all have the shortest connecting rod ratios of 1.77:1. This increases bottom end torque and explains the off-the-line response of these engines.

TC96 Connecting Rod Ratio Calculation

Example; TC96 has a con rod length of 7.762” and a stroke of 4.375”

The connecting rod ratio is the connecting rod length divided by the engine’s stroke.

Therefore;

$$\text{Con Rod Ratio} = \text{Con rod length} \div \text{Engine stroke}$$

$$\text{TC96 con rod ratio} = 7.762 \div 4.375 = 1.77{:}1$$

Connecting Rod Ratio Chart

Displacement is rounded off and approximate for popular use; for example, the Twin Cam TC96 where 96 cubic inches is not really 1575cc but 1573.44cc. However, a TC96 is not 96ci but really 96.64 cubic inches x 16.39cc's per cubic inch = 1583.93cc. How did I arrive at an accurate 96.64 cubic inch engine displacement for the TC96? The formula for

calculating cubic inches is; X 0.7854 X stroke X number of cylinders. Therefore a Twin Cam 96 = 3.750 x 3.750 bore2 x .7854 x 4.375 stroke x 2 cylinders =96.64 cubic inches.

## Why the TC96 is faster than the TC88

Many of you are going to think, what a dumb title. Of course, the bigger TC96 engine is faster than the smaller TC88. I will tell you what; there was much confusion in the winter and spring of 2006-2007 because performance people were hopping up their TC96 Twin Cams and the power their experience said should be forthcoming was not all there. The same deal happened with the Twin Cam 88 in 1998-1999.

First, neither is as fast as it could be with gear driven cams instead of the timing chains because of cam timing variances from loose sprocket/timing chain fit.

Spintron Dynamometer testing has proven that timing varies by up to 4-degrees. Secondly, the spring loaded timing chain adjusters in use from 1999 to 2006 allow an extra 4-degrees cam timing variance when using performance valve springs.

The TC96 supercharged gear ratios need to coordinate with performance work in specific ways to gain the power increases expected.

Connecting Rod Length Relative to Stroke

The TC88 and TC96 have the same length connecting rods but the stroke changes a full ⅜” from the TC88 stroke of 4.000” to the TC96 stroke of 4.375”.

This results in significant differing connecting rod ratios.

The TC88 has a con rod ratio of 1.94:1

The TC96 has a con rod ratio of 1.77:1.

This is a great and timely example of increasing the stroke on the TC96 from the TC88 relative to connecting rod length thereby decreasing the connecting rod ratio.

So what does this mean?

This is one major reason the TC96 is so fast off the line with a supercharged primary gear ratio and smaller rear pulley that carries back into the final gear ratio!

The lower connecting rod ratio of the TC96 further lowers the power band.

This is significant.

The wild increases in primary sprocket sizing in the 2006 TC88 Dyna and 2007 and later TC96 models over previous TC88 Twin Cams are remarkable. The front sprocket jumps from formerly tall 25-teeth up to a stunning tall 34-teeth! The clutch sprocket jumps from former 36-teeth up to amazing super big 46-teeth.

Furthermore, the rear pulley drops 4-nubs to 66-nubs from the previous Bagger standard of 70-nubs (It goes to 68-nubs in 2009).

If the connecting rod ratio had not decreased, the TC96 would not be so fast off the line. The decrease in TC96 rod ratio over the TC88 simply lowers the power band rpm and

thus torque into the low ranges offsetting the supercharged gear ratios! Furthermore, this is also, why there is not much expected extra horsepower forthcoming from the TC96 off-the-line even though there is an additional eight cubic inches. Remember horsepower is a multiple of torque and rpm.

The torque increases but the rpm is obviously very low off the line.

Sprocket/pulley Sizes

Primary: Secondary and Final Drive Ratios

The reason that 06 Dynas and 2007 Big Twins have differing gear ratios is because of a 2-tooth/nub reduction in the 07-rear wheel pulley. 2009 ratios are the same as the 2006 Dyna since both use a 68-nub rear pulley.

## Connecting Rod Ratio Affects Engine Wear and Longevity

A smaller or lower con rod ratio will exacerbate piston-to-cylinder engine wear because it exerts more pressure on the piston and cylinder thrust surfaces front and rear. Therefore, a taller or longer con rod ratio will be kinder to the thrust surfaces adding longevity to servicing and repair intervals.

Whoa there old buddy, the quick-to-wear Ironhead XL's have a tall con rod ratio while the relatively extremely reliable Twin Cam has a high-wear lower con rod ratio. Something is not jiving here. Well, Ironheads use the highest wear pistons while the Twin Cams utilize the superb piston construction. I describe this piston construction at length in Volume I: The Twin Cam, Chapter I: March 1998: The Introduction of the Twin Cam, subsection, Dissecting the Twin

Cam 88 Fathead under Hypo, Hyper, and Eutectic Pistons.

Twin Cams also ceramic coat the piston skirt thrust surfaces to provide a slippery and durable finish. Ring construction and quality improves by light years from the early Ironheads to these most modern of the Big Twins. Cylinder rigidity and shorter construction in the Twin Cam barrels provide a supportive base for the pistons to travel in while the Ironhead cylinders flex. Twin Cam cylinders are also rounder than the Ironheads because of the use of torque plates when resizing the cylinder bores to simulate actual running conditions.

Many respected engine builders feel that the ideal con rod ratio centers around 1.75:1 with a caveat.

The Ideal Connecting Rod Ratio

The ideal rod ratio of around 1.75:1 is only best if engine breathing is optimal for the engine design.

The late 80-cubic inch Shovelheads from 1977½ to 1984, and the Evolution engine from 1984 to 1999 have the ideal ratio of exactly 1.75:1. The late Shovels were the best Big Twin engines to date. It was better than the 1200cc/74ci early Shovels, much better than the 1948-1965 Panheads and eminently better than the 1936 to 1947 Knuckleheads.

I love all these engines. I am only talking about technological advancements here, so please do not git yer knickers in an uproar if you feel I am insulting your pride and joy.

The 1977½ to 1984 heads for the 1340cc Shovels breathing was much better than the 1200cc Shovels but still a little too much for the engine size.

I feel that the Evo breathing capabilities match better than what went before and what came after. In other words, breathing engine capability optimally matches the Evo engine design.

The TC96 is very close at to the ideal ratio at 1.77:1. However, breathing capability for engine design is lacking. I attribute this not to any fault of Harley-Davidson but to the requirements of meeting EPA and CARB emissions dictates where controlling pollutants takes priority over engine efficiency and performance concerns.

Connecting Rod Angle

The cylinders on any engine including Harley-Davidson are stationary. However, the connecting rods are not as they

rotate around every engine revolution, changing angle constantly in relation to the cylinders.

Of course, the con rods connect to the pistons, push them up, and pull them down the enclosed cylinders. The only time, the connecting rod and piston form a straight line is at TDC in the cylinder. Otherwise the connecting rod is either pushing the piston against the front of the cylinder thrust wall surface upwards on either the compression or exhaust strokes or pulling the piston down against the rear thrust surface particularly on the intake stroke.

If the connecting rod and the piston formed a straight line up and down on all four strokes, which is impossible, there would be no thrusting of the piston against the cylinder walls. The question now becomes the angle this thrusting occurs at because the greater the angle of con rod relative to piston, the greater the thrust against either front or rear cylinder wall. A greater angle occurs with lower con rod ratios, which

exacerbates with a shorter rod and/or a longer stroke.

The following conditions will occur with too short a connecting rod combining with too long a stroke creating a low con rod ratio with an adverse mechanical which means the angle of the connecting rod relative to its attaching piston is severe.

1. There is excessive friction or thrust against the cylinder wall by the piston skirt.

2. The piston length, measuring from the center of the wrist pin hole to the bottom of the piston skirt will be shorter leading once again to premature wear because of lack of support.

3. There will be small increases in engine operating

temperatures as well as oil temperature due to the increase in friction that occurs. Higher oil temperatures cause oil to oxidize and thus break down sooner.

4. Minor losses to overall horsepower will occur due to internal frictional power losses.

5. Detonation is less of an issue because of lowering combustion pressure, which leads to a cooler running engine.

6. Turbocharging, Supercharging and/or Nitrous will benefit from a shorter rod ratio because of the lowering of combustion pressure.

7. Spark advance must be greater because the piston is farther from TDC than with an engine with a more conservative con rod ratio.

8. Engine breathing, the base of all power, in terms of rising intake vacuum occurring rapidly after top dead center (ATDC), which will allow the use of bigger EFI or carburetion induction with no response loss. However, TC88 and TC96 intake ports will suffer breathing problems at a lower rpm. This occurs because of poor port design.

9. The Twin Cam engine will become less receptive to late intake valve closing when using hotter performance cams.

10. Finally, vibration will typically increase because of excessive angularity of rods relative to the pistons they drive.

## Comparing Con Rod Ratios with Intake Breathing Quality

As I wrote earlier, shorter connecting rods relative to the stroke produce low connecting rod ratios of 1.40:1 to 1.75:1

and longer connecting rods in relation to the stroke generate high connecting rod ratios of 1.75:1 to 2.20:1

*Single plenum manifolds have one manifold feeding two cylinders. This type is in use on stock Harley's. Dual plenum manifolds have a separate runner feeding each cylinder independent of the other.

Piston Movement and Position Relative to Connecting Rod Ratio

Piston speed varies depending on location during the engine strokes. I always made the assumption this was true but felt speed was changing leading up to top dead center and afterwards but made maximum speed in the middle of the stroke, speeding up to it and slowing down afterwards. I also thought this speed differential would be equal during piston axis change at bottom dead center.

I was wrong on some counts. Well, piston speed is faster at before and after TDC than it is at BDC. Furthermore, the difference in speed between TDC and BDC is not constant because it varies with the connecting rod ratio.

This sounds complex but it will make sense after the next few sentences. For example, two engines with the same stroke but differing con rod lengths will have their pistons in different locations even though the specific location of flywheel rotation is equal. How can the pistons from each engine be in the same place? The stroke is the same in both but one has a longer connecting rod than the other does. Therefore, the piston is further up the bore in the engine with the longer connecting rod.

There are two exceptions or locations where they will be at the same spot, which are TDC and BDC because the piston and rods are straight up and down.

Remember the strokes are the same. However, the shorter con-rod-ratio engine will move the piston closer to bottom

dead center (BDC), than the longer con-rod-ratio engine anytime between before bottom dead center (BBDC) and after bottom dead center (ABDC). Conversely, the longer con-rod-ratio engine will move the piston closer to top dead center (TDC), than the shorter con-rod-ratio engine anytime between before top dead center (BTDC) and after top dead center (ATDC).

Obviously, piston speed varies between the two comparative engines with the same stroke but differing con rod lengths also.

Let us look at the compression stroke and the effect piston speed will have on engine breathing. The shorter con-rod-ratio engine will have slower piston movement upwards from BDC on the compression stroke. Because of the slower piston speed, more mixture will trap and thus compress at the same point of intake valve closure when comparing with the longer con-rod-ratio engine. This event makes the shorter con-rod-

ratio engine more tolerant of a later intake valve closing. Therefore, this intake cam timing is more acceptable with a shorter con-rod-ratio engine.

In closing, the engine connecting rod ratio of con rod length relative to engine stroke to some extent will control the amount of trapped air/gas mixture, its compression, and its burning at a specific point of closing of the intake valve.

This is a reason that a shorter con rod ratio will lower the power band producing more torque at a lower rpm.

Why the Ironhead XL was faster than the Big Twin Shovel

The 1200cc/74c.i Shovelhead FL has shorter con rod ratio of 1.87:1 while the 1000cc Ironhead XL has slightly higher ratio of 1.95:1

The Shovel needs the shorter con rod ratio, which lowers the torque band to move the heavier weight off the line. The hotrod XL can tolerate a slightly higher rod ratio because it is easier to move its lighter weight off the line.

The Sportsters primary and final gear ratios also ably assist this in-the-day rocket off the line relative to the heavy slug FL, King of the Highway.

How times change!

The reputed in-the-day King of the Highway FL pales in comparison with a TC96 highway powerhouse. Speeding was not an issue with the FL. However, there are few North American city streets or highways that the TC96 can traverse legally while riding in its potential comfort zone.

XL and FL Gear Ratios

Another major factor of why the 1000cc Sportster is faster off the line than its big brother the 1200cc Shovel, which has a 17% greater engine displacement is gear reduction from the engine back to the rear wheel.

Primary Gear Reduction Ratio

Compare the XL engine highway sprocket of 34-teeth to the much smaller Shovel engine off-the-line sprocket of 23-teeth. Also, compare the XL clutch off-the-line sprocket of humungous 59-teeth to the conservative all-purpose 37-teeth on the Shovel clutch sprocket.

All other factors being equal, the Ironhead XL should be fast off the line because the number of off-the-line clutch teeth

overwhelms the number of highway teeth of the engine sprocket. If both bike's clutch sprockets were equal in teeth the result would be quite the opposite with a very tall 34-tooth XL engine sprocket relative to the very low grunt 23-tooth FL engine sprocket.

The differential between the teeth on the clutch sprockets turns all this the opposite way. There is a 25-tooth differential between the XL primary sprockets relative to a smaller 14-tooth differential between the FL primary sprockets.

Determine the primary gear ratio by dividing the number of teeth on the clutch sprocket by the number of teeth on the engine sprocket.

$$\text{Primary Gear Ratio} = \frac{\text{clutch sprocket teeth}}{\text{engine sprocket teeth}}$$

1000cc XL Ironhead Primary Gear Ratio = 59/34 = 1.74

1200cc FL Shovel Primary Gear Ratio = 37/23 = 1.61

The Sportster has a primary gear ratio of 1.74 (59 ÷ 34 = 1.74), relative to the ratio in the Shovel of 1.61 (37 ÷ 23 = 1.61).

Therefore, the Sportster engine has a primary ratio of 1.74, which means the engine rotates 1.74 times for one rotation of the clutch hub and thus, one rotation of the transmission mainshaft that the hub affixes. Moreover, the Shovel engine has a primary ratio of 1.61, which means the engine rotates a lesser 1.61 times for one rotation of the clutch hub and thus, one rotation of the transmission mainshaft that the hub affixes.

Secondary Drive Gear Ratios

The secondary ratios of both bikes are more similar. Compare the XL transmission city sprocket of 21-teeth to the slightly smaller Shovel transmission off-the-line sprocket of 22-teeth. Also, compare the XL rear off-the-line sprocket of 51-teeth to the also same conservative city 51-teeth on the Shovel final rear sprocket.

For comfortable highway riding both bikes need taller gearing.

On my Shovels, I increase the engine and transmission sprockets to 23 and 24 or when on a long highway trip to 24 and 24. I lower the rear wheel sprocket to 49-teeth on 1973 and later Shovels. If I ride an Ironhead XL, there is only one-sprocket that is amenable to change; it is the transmission

sprocket, which I increase to 23 or 24-teeth.

Hence, the colloquialism Milwaukee Vibrator suitably describes the Ironhead XL.

In my opinion, the Factory did not even start getting the gearing right on the Sportster to around 1991. By 2004, the Sportster could no longer claim title to the Milwaukee Vibrator.

All other factors being equal, the Ironhead XL should be even faster off the line because the transmission sprocket is one-tooth smaller than the FL. The FL transmission sprocket is still small in the scheme of things allowing the beast to traverse at low rpm off the line.

Determine the secondary gear ratio by multiplying the rear

sprocket teeth times the high transmission gear ratio and dividing this by the number of teeth on the transmission sprocket.

Secondary Drive Gear Ratio

$$\text{Secondary gear ratio} = \frac{\text{wheel sprocket teeth x high gear ratio}}{\text{Transmission sprocket teeth}}$$

FL 74 Shovel Secondary gear ratio $= 51 \times 1 \div 22 = 2.32{:}1$

Ironhead 1000 XL Secondary gear ratio $= 51 \times 1 \div 21 =$ 2.43:1

Both fourth and final high gears in the Sportster and the Shovel are a 1:1 ratio.

The Sportster has a secondary gear ratio of 2.43 (51 x 1 ÷ 21 = 2.43:1), relative to the ratio in the Shovel of 2.32:1 (51 x 1 ÷ 22 = 2.32:1). Therefore, the Sportster rear drive has a secondary ratio of 2.43:1, which means the transmission sprocket and transmission mainshaft rotate 2.43 times for one rotation of the rear wheel sprocket and thus the rear wheel.

Moreover, the Shovel rear drive has a secondary ratio of 2.32:1, which means the transmission sprocket and transmission mainshaft rotate 2.32 times for one rotation of the rear wheel sprocket and thus the rear wheel.

Final Drive Gear Ratio

The primary and secondary ratios determine the final drive gear ratio by multiplying the two together.

Final Drive Gear Ratio = Primary Ratio x Secondary Ratio

FL 74 Shovel final gear ratio = 1.61 x 2.32 = 3.74:1

Ironhead 1000 XL final gear ratio = 1.74 x 2.43 = 4.23:1

The Sportster has a final drive gear ratio of 4.23:1 (1.74 x 2.43 = 4.23:1), relative to the ratio in the Shovel of 3.74:1 (1.61 x 2.32 = 3.74:1). Therefore, the Sportster final drive has a gear ratio of 4.23:1, which means the engine rotates 4.23 times for one rotation of the rear wheel sprocket and thus the rear wheel.

Moreover, the Shovel final drive has a gear ratio of 3.74:1, which means the engine rotates 3.74 times for one rotation of the rear wheel sprocket and thus the rear wheel.

Finally, the Sportster is faster off the line rotating the engine 4.23 times for every turn of the rear wheel while the Shovel engine rotates a lesser 3.74 times per rear wheel revolution.

Another obvious answer is the weight-to-power ratio of the lighter Sportster to the heavier Shovelhead. Yet the Sportster was the hotrod Harley of the day even with 200 cc's less engine displacement. The Sportster had 1000cc while the early Shovel boasted 1200cc.

So what gives? An obvious and speculative answer might be the con rod ratios accomplish this feat. However, the connecting rod ratio is higher, not lower on the XL. This means the power band moves up the rpm scale and does not create as much low-end torque.

A lower con rod ratio, which could overcome the supercharged gear ratio, is not the reason the Sportster is faster. The Shovel, as a whole was bigger and heavier and needed lower gear ratios just to get moving. The Sportster, as a whole was lighter and more nimble so it could accommodate higher gear ratios.

Thus, the Sportster had a favorable weight-to-power ratio. It creates more power relative to its weight than the big Shovel Bagger.

How did Harley increase the speed of the Shovel? The Motor Company created the FX series, which is a stripped down Bagger FL, thus increasing the weight-to-power ratio by lowering the FX weight. FX gear ratios could also then modify.

Another factor helps the Sportster against its clunky and awkward big brother.

Internal valve train weight is also lighter. This allows for more responsive and higher revving capabilities because there is not as much mechanical weight to push with all the limitations of more inertia and higher momentum physics with parts that must speed up, slow down, stop, reverse direction and repeat this thousands of times per minute.

One final factor is that more torque creates with the Sportster, not with a smaller con rod ratio but with an increase in compression ratio over the Shovel. Increased compression creates low-end torque to offset the higher gear ratios and the taller con rod ratio.

So there we have it. The Ironhead Sportster should be

slower because of tall gear ratios and a taller con rod ratio. However, it is faster, overcoming these low-end power negatives by a favorable weight-to-power ratio, higher compression and a lighter valve train. In fact, the Sportster can rev higher and get there faster because of the supposedly negative factors.

Many factors intertwine to create a faster Harley. For those riders not-in-the-know, these factors can also combine to make a Harley much slower after spending much money to make it

## Power-to-weight and Weight-to-power Ratios

Easy understanding of ratio occurs by expressing engine horsepower divided by the engine's weight. Power-to-weight is useful in comparing an engine's power with another.

Do not confuse or think I have made a typographical error when I use the terminology power-to-weight ratio as opposed to weight-to-power ratio. The two terms do not mean the same thing although they express similar concepts.

Power Loading also known as Weight-to-Power Ratio expresses as the engine horsepower divided by the motorcycle running weight including the rider's weight. Weight-to-power ratios take into account that one bike may be heavier than another and thus affects acceleration.

Many vehicle manuals will express the dry weight of the vehicle, which in this case is the weight of the Harley-Davidson itself without gas or rider. Motorcycle running weight is a Harley ready to ride without rider. The fastest Harley will be the one that has the most power relative to its weight.

Sir Isaac Newton's Second Law of Motion states that acceleration is proportional to the motive force divided by the mass of the object. In other words, a Harley's acceleration is comparative to horsepower divided by the combined weight of the bike and rider.

$$\text{Power-to-weight ratio} = \text{power} \div \text{engine weight}$$

This formula is one-dimensional in that it is only useful for comparing one engine to another before added factors of chassis and rider weight, gear ratios and transference loss of horsepower down the powertrain.

$$\text{Weight-to-power} = \text{power} \div \text{total weight}$$

Now, this is simplistic, also, because the equation does not factor in how the force or horsepower applies to the chassis.

By this, I mean that other factors like dissipating loss of power down the powertrain from the engine as well as the all-important gear ratios do not include in the equation.

Therefore, weight-to-power ratios are overly optimistic when only dividing engine power by the gross weight of the bike and rider.

Power-to weight ratios may also apply to comparing engine weight with its horsepower output to another engine's horsepower divided by its weight.

I will compare the power-to-weight ratios of an 883 and 1200 Sportster. First, I need to calculate the horsepower of each since Harley-Davidson only gives torque figures. As, the reader knows the calculation of horsepower obtains by dividing rpm by the mathematical constant 5252 and

multiplying this by the torque.

The Sportster 883 produces about 55 ft-lbs of torque at 3500 rpm (3500 rpm ÷ 5252 x 55 ft-lbs = 36.65 horsepower) with an educated engine weight guess of about 185 lbs. To answer the obvious, yes I have triple-checked the horsepower generated by the 883 engines. The 883 engine weighs more than a 1200 XL engine because the cylinders bore out a full .498" to achieve to extra cubic inch displacement. The ½" material weighs about 6-lbs; therefore, a 1200cc XL engine weighs about 180 lbs. The exact weight is unimportant for the discussion whose goal is demonstrating a power-to-weight ratio comparison.

The Sportster 1200 produces about 79 ft-lbs of torque at 4000 rpm (4000 rpm ÷ 5252 x 79 ft-lbs = 60.16 horsepower). Now, the reader can already tell that the 883 is a dog in the power department and the 1200 is relatively fast, especially with a light chassis weight.

Comparing Power-to-weight Ratios

XL 883 and XL 1200 cc Sportsters

*Approximate educated guess

In the example, the relative horsepower calculation figures skew because maximum torque is at a different rpm for each bike. In reality, the power is closer with the 1200 having more because of bigger displacement. The 883 XL engine has a very lower power-to-weight ratio of .20 compared with the 1200 XL engine with a very much higher ratio of .33. The 883 Sportster produces a measly .20 horsepower per pound of engine weight while the 1200 Sportster produces a more

respectable .33 horsepower per pound of engine weight.

The power loading or weight-to-power ratio exaggerates even more. Suppose we eliminate rider weight as a varying factor by assuming a rider of equal weight is riding both the 883 and 1200 Harley's. The running weight of the 1200 XL is about 20 lbs less than the 883 XL. This factor will make the 1200 even faster because of less weight over the 883 Sportster.

Power Loading: Weight-to-power Ratios

However, weight-to-power ratio is a common methodology to compare one engine's power relative to the weight it must push to another. This is why I can say that a Dyna with a 150 lb rider will have a higher weight-to-power ratio than an FLHT with a 250 lb rider.

In the example, the relative horsepower calculation figures skew because maximum torque is at a different rpm for each bike. In reality, the power is closer but still not equal with the Bagger having slightly more because of bigger mufflers.

I can say that the Dyna with a lighter rider will be much faster than an FLHT with a heavier rider since the primary, internal transmission, secondary and final gear ratios are the same.

## Comparing Weight-to-power Ratios

### Power Loading

The TC96 FLHR Road King has 61.7 HP, (3500 rpm ÷ 5252 x 92.6 ft-lbs of torque = 61.7 HP) + a running weight

of 777 lbs + rider weight of 250 lbs =1027 lbs.

The TC96 FXD Dyna has 52.6 HP, (3000 rpm ÷ 5252 x 92 ft-lbs of torque = 52.6 HP) + running weight of 663 lbs + rider weight of 150 lbs = 813 lbs.

FLHR Weight-to-power = power ÷ weight

FLHR = 61.7 ÷ 1027

FLHR = .060

FXD Weight-to-power = power ÷ weight

FXD = 52.6 ÷ 813

FXD = .065

Therefore, even though the FXD Dyna example has lower horsepower, the lesser weight of bike and rider easily overcomes this handicap to produce a higher weight-to-power of .065 versus the FLHR .060. Okay, now what does these weight-to-power ratios mean? In the English measuring system, the FLHR produces .060 horsepower per pound (hp/lb), while the FXD produces .065 hp/lb.

In the metric system, these production figures measure in horsepower per kilogram (hp/kg). A kilogram weighs 2.2 lbs. The reader may also see these ratios measure in watts per horsepower (watts/hp) or kilowatts per horsepower (kw/hp). One horsepower equals 745.7 watts. One horsepower equals .7457

Why the Sportster is faster than a TC96

I realize that some Big Twin riders look down at the lowly Sportster as an inferior machine because of its smaller size. The Sportster has always been faster than the Big Twin has although many do not think this because of the smaller engine versus a bigger one.

Ahh, it is time for another one of Donny's Blasts from the

Donny's Blasts from the The Mild-mannered Pharmacist

In the nineteen-nineties, at Heavy Duty Cycles, (www.heavydutycycles.com), I had the pleasure of working on a Sportster owned by a seemingly mild-mannered pharmacist replete with eyeglasses who feigned shyness. He loved his Sportster. It took me awhile to get onto David, as his act was a study in sucker hustling. This guy could teach Fast

Eddie Felson, the pool shark, a lesson or two.

In Toronto, macho Big Twin riders derisively referred to the Sportster as a girl's bike or as a sissy Harley. When people were watching, David became a nervous Nellie, awkwardly riding his pride and joy.

David had me increase the displacement to 1200cc, do some headwork, install a set of performance cams with some carburetor work and change the gearing a little. Nothing was overt; the changes were discreet and unnoticeable except to the most discerning of eyes.

If seeking a victim, David would frequent dealership get-togethers, Hog events and such, patiently waiting for the inevitable. Invariably, some buffoon on chrome Softail or tricked out Bagger would start the insulting side comments about Sportsters. David, politely and shyly feigned

subservience to these self-important oafs. He would ask them innocuous questions and channel them into boastful bouts of dialogue about how fast their bikes were.

Of course, this was in front of the guy's friends and watchers-on especially the girls. Soon, David would lead the cock-a-doodle-doo Big Twin rider into the trap by timidly saying that he thought his Sportster was pretty fast.

Anyway, David was much better at hustling than I am writing about it. The race would be on for a small wager. Most times David would just beat the Big Twin by a little, enough to win the bet but always leaving doubt for the next time. The next time usually came within minutes as David supplemented his already excellent income.

However, if the guy was particularly insulting, David

would wait until he was alone, sometimes following him away from the event. Then David would issue an authoritative challenge and do a wheelie whilst cleaning the Evo Big Twin's clock.

The same holds true today even though the Sportster engine has no displacement increase while the TC96 is 16 cubic inches bigger than the 80 cubic inch Evolution Big Twin. Therefore, if ever wondering why a 1200 XL can beat a TC96 even though the cubic inch displacement of the XL is 74 cubic inches relative to the size of the ninety-six cubic inch TC96, we need look no further than weight-to-power ratios.

This full 22 cubic inches bigger displacement cannot offset the weight differential of the 1200 XL, which weighs in at about 562 lbs relative to the TC96 FLHR at 777 lbs; look no further than the weight differentials of the complete motorcycles. I can easily prove this by comparing weight-to-power ratios of the two Harley's. These mathematical

assessments are known as power loading comparisons.

## Comparing Power Loading Weight-to-power Ratios

### 1200 Sportster versus the TC96 FLHR Road King

The 1200 XL ratio produces .081 horsepower per pound of weight (60.16 ÷ 742 = .081).

The TC96 FLHR ratio produces .064 horsepower per pound of weight (61.7 ÷ 957 = .064).

Therefore, the smaller displacement 1200 Sportster with 22 less cubic inches produces .081 horsepower per pound while the bigger TC96 engine manages just .064 horsepower

per pound. Of course, there are other differentiating factors like differing gear ratios. These help equalize power, particularly at higher speeds.

## Power Formulae

I will give the formulae and examples for the following,

1. Calculate horsepower from quarter-mile speed and weight;

2. Calculate horsepower from elapsed time (ET) and weight;

3. Calculate elapsed time from horsepower and weight;

4. Calculate quarter-mile speed from horsepower and weight;

5. Calculate quarter-mile speed from elapsed time and

6. Calculate elapsed time from quarter-mile speed.

## Quarter-Mile Horsepower

The reader will see a practical example of power-to-weight ratio. The first example will have an 800 lb FLHT and rider with a ¼-mile speed of 100 mph, which is a good speed for an 800 lb bike/rider. The second example will be a 700 lb Dyna and rider with a ¼-mile speed is 108 mph, which is realistic for some mild performance work. Engine horsepower for the Twin Cam Dyna and the FLHT rate very close by Harley-Davidson. I have taken reasonable weights and elapsed times for the two bikes. The FLHT makes a respectable 72.16

horsepower while the Dyna increases this to 79.51

horsepower at the rear wheel.

Calculate Horsepower from Quarter-mile Speed and Weight

Weight is total of Harley + rider

The figure 223 is a constant

Example 1: FLHT

Example; ¼-mile speed is 100mph; rider + H.D is 800 lbs

HP = (mph ÷ x weight

HP = (100 ÷ x 800

HP = (.4484) 3 x 800

HP = .0902 x 800

HP = 72.16

Example 2: Dyna

Example; ET = 13 seconds; rider + H.D. is 700 lbs

HP = (mph ÷ x weight

HP = (108 ÷ x 700

HP = (.4843) 3 x 700

HP = .1136 x 700

HP = 79.51

Elapsed Time (ET) Horsepower

The reader will be able to see just how important weight to power really is.

The first example will have an 800 lb FLHT and rider with an elapsed time of 14 seconds, which is a good speed for an 800 lb bike/rider. The second example will be a 700 lb Dyna and rider with an elapsed time of 13 seconds, which is realistic. Horsepower for the Twin Cam Dyna and the FLHT rate very close by Harley-Davidson. I have taken reasonable weights and elapsed times for the two bikes. The FLHT makes only 43.7 horsepower while the Dyna has 66.1 horsepower.

Calculate Horsepower from Elapsed Time and Weight

Weight is total of Harley + rider

ET = elapsed time

The figure 5.92 is a constant.

A number cubed is that number times itself times itself. For example, 4 cubed, normally written is $4 \times 4 \times 4 = 64$.

Example 1: FLHT

Example; ET = 14 seconds; rider + H.D. is 800 lbs

HP = weight ÷ (ET ÷

HP = 800 ÷ (14 ÷

HP = 800 ÷

HP = 800 ÷ 18.30

HP = 43.7

Example 2: Dyna

Example; ET = 13 seconds; rider + H.D. is 700 lbs

HP = weight ÷ (ET ÷

HP = 700 ÷ (13 ÷

$$HP = 700 \div$$

$$HP = 700 \div 10.59$$

$$HP = 66.1$$

## Calculating Elapsed Time (ET)

I will demonstrate how important weight is to shaving time in a race. The first example will have an 800 lb FLHT and rider with 65 horsepower, which results in an elapsed race time of 13.67 seconds. The second example will be a 700 lb Dyna and rider with 65 horsepower, which results in an elapsed race time of 13.08 seconds. This difference in elapsed time is full .59 seconds, which is significant. The reader can calculate via other formulae presented here the increase in horsepower necessary to cut over a half-second off the race

time.

Elapsed Time from Horsepower and Weight

Weight is total of Harley + rider

ET = elapsed time

The figure 5.92 is a constant.

Example 1: FLHT

Example; HP = 65; rider + H.D. is 800 lbs

ET = 5.92 x (weight ÷ horsepower)3

$$ET = 5.92 \times (800 \div 65)\ 3$$

$$ET = 5.92 \times (12.3)\ 3$$

$$ET = 5.92 \times 2.31 = 13.67$$

$$ET = 13.67 \text{ seconds}$$

Example 2: Dyna

Example; HP = 65; rider + H.D. is 700 lbs

$$ET = 5.92 \times (\text{weight} \div \text{horsepower})3$$

$$ET = 5.92 \times (700 \div 65)\ 3$$

$$ET = 5.92 \text{ x } (10.8)\ 3$$

$$ET = 5.92 \text{ x } 2.21 = 13.67$$

$$ET = 13.08 \text{ seconds}$$

Calculating Quarter-Mile Speed

I will demonstrate that less weight wins the race. The first example will have an 800 lb FLHT and rider with 65 horsepower, which results in a quarter-mile speed of 96.78 mph. The second example will be a 700 lb Dyna and rider with 65 horsepower, which results in a quarter-mile speed of 100.35 This difference by lowering weight from 800 lbs to 700 lbs in quarter-mile speed is 3.57 mph.

Therefore, all other factors being equal, less weight wins the race.

Quarter-mile Speed from Horsepower and Weight

Weight is total of Harley + rider

The figure 223 is a constant.

Formula: MPH = 223 x (horsepower ÷ weight)3

Example 1: FLHT

Example; HP = 65; rider + H.D. is 800 lbs

MPH = 223 x (horsepower ÷ weight)3

$$MPH = 223 \times (65 \div 800)3$$

$$MPH = 223 \times (.08125)3$$

$$MPH = 223 \times .434$$

$$MPH = 96.78$$

Example 2: Dyna

Example; HP = 65; rider + H.D. is 700 lbs

$$MPH = 223 \times (\text{horsepower} \div \text{weight})3$$

$$MPH = 223 \times (65 \div 700)3$$

$$MPH = 223 \times (.0929)3$$

$$MPH = 223 \times .450$$

$$MPH = 100.35$$

Calculating Quarter-Mile Speed from Elapsed Time

Weight has a significant influence on elapsed time. Therefore, a faster elapsed time will result in higher quarter-mile speed. The first example will have an FLHT with an elapsed quarter-mile time of 14-seconds, which results in a quarter-mile speed of 94.29 mph. The second example will be a Dyna with an elapsed quarter-mile time of 13-seconds, which results in a quarter-mile speed of 101.54 mph. The difference in a reduction of one second from 14-seconds to 13-seconds elapsed times will result in a significant increase in

quarter-mile mph of 7.25 mph.

Quarter-mile Speed from Elapsed Time

ET = elapsed time

The figure 1320 is a constant

Formula: MPH = (1320 ÷ ET)

Example 1: FLHT

Example; ET = 14 second quarter-mile

MPH = (1320 ÷ ET)

$$MPH = (1320 \div 14)$$

$$MPH = 94.29$$

Example 2: Dyna

Example; ET = 13 second quarter-mile

$$MPH = (1320 \div ET)$$

$$MPH = (1320 \div 13)$$

$$MPH = 101.54$$

Calculating Elapsed Time from Quarter-Mile Speed

Quarter-mile speed has a significant influence on elapsed time.

Therefore, a faster quarter-mile speed will result in a faster elapsed time.

The first example will have an FLHT with a quarter-mile speed of 90 which results in an elapsed time of 14.7 seconds.

The second example will be a Dyna with a quarter-mile speed of 100 mph, which results in an elapsed time of 13.2 seconds.

The difference is a noteworthy reduction in elapsed times of 1.5 seconds from 14.7-seconds to 13.2-seconds.

Elapsed Time from Quarter-Mile Speed

$$ET = \text{elapsed time}$$

The figure 1320 is a constant

$$\text{Formula: } ET = (1320 \div \text{mph})$$

Example 1: FLHT

Example; 90 mph quarter-mile

$$ET = (1320 \div \text{mph})$$

$$ET = (1320 \div 90)$$

ET = 14.7 seconds

Example 2: Dyna

Example; 100 mph quarter-mile

ET = (1320 ÷ mph)

ET = (1320 ÷ 100)

ET = 13.2 seconds

Lies, Lies, and Dynos

I apologize in advance to all the legitimate dyno people out there that are a credit to the tuning and hi-performance

industry.

Without skilled Dynamometer and Flow Bench people, how would the consumer be able to judge a product? More importantly, how would manufacturers be able to gauge their research and the directions it should take for further refining and improvements that make our Harley's faster?

I use this title to both attract readership and to explain the necessity of caveat emptor or buyer beware when presented with seemingly irrefutable proof concerning a product. Some unscrupulous Dynamometer (Dyno) operators or those without enough knowledge to interpret readings will use smoke-and-mirror tactics just like the snake oil salespersons of old to demonstrate their products work the best and are superior to all the others in the marketplace.

The Lies

There are two types of lies basing on unscrupulous dyno work.

First, it is not the dyno that lies but the operator. Even if rigging a dyno to give false readings and graphs, the operator set it up that way. I will go through unscrupulous operator slight-of-hand techniques throughout this subsection.

The bigger issue since it affects more riders is dyno-based claims that one product is superior to competitor products. Surely rigged testing can illegitimately prove this but even if legitimate, often critical information is missing that explains away why one dyno test is superior to another.

Let us look at cams, exhausts, carburetors, manifolds, valves, heads…you name it, where testing manipulates to

demonstrate superiority. How can three different cams made by three different manufacturers be the best? It is impossible or is it? Advertising purports these truths all the time. The claims are everywhere. However, these claims serve only to confuse the consumer creating mistrust. Capitalism is not perfect since only a product's positives reach the consumer ear while negatives to the same product lay buried.

Many times the lie is the failure to disclose all relevant information. The advertisers do themselves a disservice by carelessly assuming that the Harley rider does not notice this.

When I started riding and attempting to learn via reading magazine tech articles it was a real joke. Not only did I not know what I was doing but also neither did the magazine technical writers of the day. The blind were leading the blind. It is a different ballgame now. The consumer is increasingly

educated.

The truth is that one product can be the best if given all the relevant data! Okay, now say the advertisers are all claiming their camshaft(s) is the best and they all publish dyno graphs to prove their claims. In addition, they add another layer of credibility by using a separate non-affiliated and well-respected dyno company in the industry. When this happens, as it seems to with unnerving frequency, it only demonstrates lack of pertinent information disclosure.

There is a simple explanation. One cam may produce higher gains with one exhaust than another may and the same is very well true for other parts such as carburetors, pushrod kits, valve springs, and ignition systems. Moreover, the difference may lie in a simple tuning aid such as re-jetting a carburetor or downloading a superior map into an EFI electronic control module.

As I will demonstrate further on, the dyno result may legitimately claim more peak horsepower. However, peak horsepower means little on the street where midrange torque will rule. In fact, peak horsepower may come at huge expense by robbing torque during midrange rpms!

The riding consumer need to know these points in order to make an educated decision but even then, comparisons will be difficult since the dyno runs will not reflect exactly which parts are in your bike or what you want to purchase for your ride. Furthermore, the dyno run testing may be on a different Harley model from yours. A Softail produces different power from a Heritage, an FLHT, an FXR, a Dyna and especially a Sportster and they have different power bands but more on this later.

Different power-to-weight ratios produce different power. In fact, differing weight bikes need different cam profiles period!

Gear ratios and transference of horsepower through more efficient powertrains also will alter figures.

Real world power will subvert on an inefficient

aerodynamic FLHT relative to say, the Evo FXRT whose fiberglass design allowed faster speeds.

I have not even gone into many other dyno-related factors yet that could make comparisons difficult at best between different dynos in different locales under different conditions. With all this said and done, general comparisons are possible, making allowances for the differences as the reader grows to understand potential pitfalls.

The next area to look at is with the rear wheel dyno from which most relevant readings derive. The engine may have much performance work but power is lost through the powertrain. Normally, this will range between 10% on a properly tuned motorcycle, to as much as a disgraceful 18% on a mistuned bike with power-wasting deficiencies. These may include improper tire pressure, misalignment, worn

chains or belts and/or a slipping clutch.

A properly tuned motorcycle with everything adjusted correctly is probably a given with professional dyno testing but what if one manufacturer is using stock component parts while the other puts in an aftermarket clutch with 25% more surface area, shocks that are far superior to stock and a super sticky rear tire. These sneaky aspects may not reveal in the advertisement. However, they are effective constituents in the war of hi-performance parts sales.

Engine or Rear Wheel Effects on Dyno Readings

The next comparison is an obvious one. The Factory does their published horsepower readings on an engine dynamometer not a rear wheel one. The engine dynamometer will give far higher readings to an uneducated public. The manual may say, for example, that a particular year Evo

Softail produces 69-horsepower. This is at the engine shaft and not the rear wheel. I have tested many Softails that run fine but only produce around 45-ponies at the rear wheel.

An engine and powertrain tune up will invariably increase these figures into the fifties range but this is still a long way away from the 69-factory figure. Therefore, ensure comparison figures from different manufacturers are from the same type of dyno.

Motorcycle Break-in Effects on Dyno Readings

Suppose one dyno reading results from a low mile Harley that has not loosened up to, its true potential but the comparative one has say 40,000 miles on her. The loose moose is going to produce a higher figure, all things being equal. The tight engine loses potential horsepower through

friction while the loose one has more potential power. I am not splitting hairs here as a rider can literally feel the increase in power that begins to occur around the ten to fifteen thousand mile marks.

There are many ways to cheat on a dyno and sometimes the operator does not even realize that an unfair comparative reading is going to result. I say comparative because what is the purpose of a dyno reading if it cannot compare to another reading whether it is for tuning, a parts performance indicator or for peak horsepower blowouts for the big dick crowd.

## Cold and Hot Tires, Inflation, and Tie-down Effects on Dyno Readings

Suppose a rear tire is over or under inflated when it ties down onto the dyno or how about if different bikes strap down harder or easier. This is a simple example of how a

reading can gain inaccuracy relative to others. The size of the compressed wheel changes through the variations listed above, which affects the gear ratio that in turn affects the power band and changes the horsepower readings.

Now, compare dyno testing of a bike before it is at a correct operating temperature. Test it again after some riding where everything is nicely warmed up including the tires, which get stickier at operating temperature. Cylinder pressure will also build from a cold engine to a hotter one. Compression and torque inexorably intertwine. Same bike, same dyno, but the second reading will increase by a few horsepower.

Compare Torque/Horsepower Graphs with Riding Style

When looking at a dyno graph compare the rpm readings

corresponding with torque and horsepower curves to your riding habits to get a feeling of how this advertised part would help or hinder your performance goals. For example if an exhaust system gives 43 horsepower at 3000 rpm but touts a whopping 69 ponies at 6500 rpm, is this appropriate to riding habits. First, a Twin Cam or any Big Twin for that matter will not rev at 6500 rpm without the valves trying to change holes. Secondly, if it would who cares?

Useable street powerband varies in the area of 2000 to 4000 rpm. Therefore, observe the power band on the graph to see if the advertised component fits in with normal riding habits.

The proper usage of a dynamometer as a diagnostic and tuning tool is much more advantageous than the fun aspect of dyno shootouts at biker parties. Whether jetting a carb, testing different air filters or elements, carburetors, cams, clutches or whatever else, the dyno through successive runs will indicate

whether performance and efficiency is improving or not. The dynamometer is a wonderful and informative tool in the right hands.

Absorption, Brake, or Engine Dynamometer

The Absorption Dynamometer is the same as the Brake or Engine Dyno, all differing descriptive for the same tool. Hence, the readings come from the output sprocket shaft of the engine and not the rear wheel.

The term brake horsepower evolves from this type of dynamometer readings.

The Absorption/Brake/Engine Dynamometer is primarily a research and development tool, which tests engine power without losses of transferring horsepower back through the

powertrain to the rear wheel.

Secondary but very important functions are as a superb tuning assist, an accurate specification verification tool and of course, engine break-in.

Manufacturers like Harley-Davidson typically publish horsepower and torque figures for their vehicles from this type of dyno measuring tool. The figures are always 10-15% higher than those that measure at the rear wheel all factors including tuning being equal. Furthermore, there are no gear ratios to enhance engine power for differing applications like riding style. Therefore, they are more impressive but not reality, where it counts at the tire patch contacting the asphalt.

The absorption dyno replicates a simulated upward plane

or hill that the operator can progressively make steeper until acceleration is no longer possible. Much heat generates with absorption dyno use but computerized automation allows for fast measuring and recording sparing damage relative to manual slow human manipulation.

There are also rear wheel dynos basing on the absorption-hill model still kicking around. They are an older type of dyno that may have computerization upgrades or operate manually.

Inertia Dynamometer

The Inertia Dynamometer became possible because of computerization. This dyno is one more familiar to most than the engine one. Inertial dynos measure an engine's ability to accelerate a heavy mass, most often a roller. The engine works against this heavy mass on a level as opposed to its motorcycle weight on the increasing steep plane. Horsepower

figures derive from the mass (weight), acceleration and speed. Torque calculates easily with our formula by multiplying the horsepower by 5252 and dividing by the rpm.

The motorcycle rear tire sits between two rollers where resistance measures and translates into useable figures. For some, tire wear is a consideration.

The calculations for horsepower and torque occur frequently allowing the dyno to print out a graph plotting the horsepower and torque curves that intersect at 5252 rpm. A typical inertia motorcycle dyno performs only wide-open throttle acceleration tests.

Heat controls more easily with an inertial dyno than the absorption one because the engine only loads for seconds. Calibration controls more easily also since there is only one moving part. In fact, the only potential calibration problem is

the dyno operator.

Chassis Dynamometer

The Chassis Dynamometer is a hybrid machine more correctly called an inertial chassis dynamometer that uses only one heavy roller that usually weighs 875 lbs. rotates and spins up to 200 mph. This is a real-life example but there are many variations. The exact weight is not a factor since every roller will have a stamped weight on it that's mass coordinates with the computer for exact readings.

Tampering with the roller drum weight will change the readings. For example, the dyno operator can obtain false higher readings by simply adding weight to the roller.

The dyno measures power by tying down with the rear

wheel on the roller. There is less resistance than with the two-roller dyno. This reduces tire wear and importantly gives more readings that are more accurate.

Eddy Current Absorber Module Dynamometer

The sophisticated Eddy Current Absorber module dynamometer applies a variable, computer-controlled load to the roller. Tests perform at differing rates of acceleration, at constant speeds at part or full throttle. Simulation of oval tracks, road courses or for our purposes high-performance street applications is a given with the eddy current absorber.

Another difference with this type of in-shop dyno is a blower fan driven by the roller that proportionally blows cooling air to the simulated motorcycle speed over the engine and radiator if there is one. Previous dynos do not introduce

any type of wind resistance. Cooling air also blows onto the tire patch connecting to the roller to combat tire deflection heat losses. Tires grow taller with heat and use (friction).

The inertial chassis dyno as well as the eddy current variation utilize the motorcycle exhaust and induction system along with ignition etc., which, allows for both a tuning tool and a performance check that increases with tuning. Many riders think that the dyno is a tuning tool for the engine only but it is in fact a whole-motorcycle tuning tool. By this, I mean that even a loose and worn chain detects on the graphing if the operator can read it correctly.

A Dynamometer is a Superlative Tuning Aid

Horsepower shootouts are purely for expensive fun. Measuring peak torque and horsepower is for machismo bragging rights but neither of these dyno functions is the

original purpose of a dynamometer.

In fact, peak horsepower achievements are not what necessarily make a fast bike although it may win the stationary contests held on a dynamometer. Mid-range torque will beat the stationary peak horsepower champion every time for two reasons. The bike will accelerate faster and longer with better mid-range torque.

Remember, once again that horsepower is merely the multiple of torque and rpm. The high rpm theoretical peak-horsepower motorcycle has a very high rpm component with a lesser amount of torque in this area of the rpm range. In short, the figure sounds great to the uninitiated but the number is meaningless to street grunt power that will win the race.

The real life useable midrange horsepower derives from a high torque component and a lesser rpm factor over about a third of the rpm range.

Secondly, the stationary peak horsepower bike will inevitably have the midrange blahs in the torque department.

How a Dyno Works

Imagine starting a Harley-Davidson in neutral and then twisting the throttle wide open and leaving it there. The engine will suffer damage possibly severe.

On a dynamometer, a load applies to the running bike, while in gear throughout the rpm range to measure the resistance the engine can handle at different engine speeds. I turn a Harley Twin Cam's throttle to say 4000 rpm on a dynamometer and apply enough of a load to keep it there

until the computer can record the resistance, which is in an instant. Then I apply additional load to reduce the engine speed down to 3,500 rpm for recording purposes. I continue to do this every 500-rpm down the range allowing the computer to record the load application.

I do this a few times to warm the engine up so that I can record the top of the power band. Once passing the peak power, there is no point in going further although tuning may bring the power band further up the rpm range or lower depending on needs and wants.

Torque and horsepower readings will probably increase with each dyno run as the engine and powertrain warm up to operational levels.

The dyno measures torque in pound-feet (ft-lbs).

Horsepower is a simple mathematical conversion whereby I, actually the computer does it automatically, can convert torque to horsepower figures by multiplying the torque by rpm and dividing this by the magic number 5252.

The most popular motorcycle dynamometer is the Dynojet (www.dynojet.com), which is an inertial type. It has a one-pulse per drum rotation sensor that sends the pulse information to an onboard computer that calculates this information into a myriad of helpful diagnostic results. Horsepower calculates mathematically using Sir Isaac Newton's physical law that states:

Force = mass x acceleration

And another law of physics that states:

Work = force x distance

Therefore, horsepower calculates from the drum weight of approximately 875 lbs, acceleration and drum velocity.

The difference between one Dynojet and another will be small weight differences of the rotating parts; drum, drum shaft, and optional braking system. To prevent dyno tampering, this rotational weight mass is encrypted and stored in the computer.

Calculating torque and the speed at which, the horsepower occurs requires engine rpm. Rpm flows from an inductive pickup in the ignition system, usually connecting to a spark plug wire.

My Dyno Advice

The reader will be in safe territory if cogniscent of the following advice.

1. Check the dyno out to see if it has an air-fuel sniffer. Do not waste your money if it does not since accurate information flows from real-time air-fuel-ratios.

2. Next, look to see if the dyno operator uses the air-fuel-sniffer since the motorcycle tuning accuracy is dependent on AFR.

Again, carbureted bikes will have a plotted AFR line that is difficult to make straight (consistent), primarily because of the inefficiency of the fuel delivery and overlapping fuel circuits relative to EFI. Fuel injected bikes should have a much straighter AFR line and something is definitely amiss if it is distorted or wavy.

3. Watch other bikes on the dyno before paying your money. Does the operator respect and show care for the tested bikes? Does he rev them carelessly past the power band?

4. Does he warm up the bikes on the dyno?

5. Is there a blower fan to cool the bikes?

6. Does he tell riders to let their bikes idle to warm them up before strapping them down on the dyno bed? Air-cooling does not work at idle. Furthermore, dyno warm ups under a cooling fan will warm up the whole powertrain also and not just the engine.

7. It is not a good practice to shock the powertrain

components. It takes at least two runs to warm up a bike and often three or four. If the operator performs dyno testing immediately, walk away.

As I have said, power will increase with each run until the bike, powertrain and tires are at operating temperatures.

8. Observe to see if the operator determines a baseline. If the operator uses the first run as a baseline, doing some tuning smoke-and-mirrors after and between each successive run, artificially higher figures will result.

It is common for a five to ten horsepower increase to occur during warm-up. A true baseline will occur after warm-up. The baseline occurs when two successive graph lines will overlay each other. Now a valid comparison base will accurately reflect power increases due to legitimate and skilful tuning efforts.

9. Obviously, a dyno operator that takes money for one or two runs is not allowing for any meaningful testing especially if ending with a comment that the bike is in good tune and the printout graph is missing the all-telling AFR line.

Okay, if meeting the previous criteria, then the dyno operator earns the opportunity to test your bike. However, still keep your eyes wide open.

10. My advice is to have the dyno operator enter your name or your bikes vehicle identification number (VIN), or some other personal preference information into the computer so that it appears on every piece of paper the computer spits out. The reason is obvious. This will ensure that the information and motorcycle tested is yours and not something else.

If the Twin Cam is stock, expect anywhere from a three to six horsepower gain from a superior free-flowing air cleaner and filter depending on brand. I have seen up to eleven horsepower from this simple and inexpensive change. If gaining only three horsepower, as welcome as the gain is, perhaps a different choice is in order. There are many excellent choices in the marketplace. I talk about some of them in Chapter IX: Fuel Delivery, subsection, Air

Mufflers or an exhaust system will accord about the same increases. There is much suppressed power to unleash in late Evos and more so in Twin Cams due to ever-increasing restrictive EPA and CARB dictates.

Dynamometer Correction Factors

Why are correcting initial dyno readings important?

The exercise is to standardize the data so that one engine or motorcycle may realistically compare to another. Therefore, the same standards must apply in the correction of dyno figures. The dyno readings must correct to a common standard or they become invalid. This is the next area for the consumer to be cognoscente of, since manipulation of printout graphing, is possible. Even, if not conspiratorial, it is necessary not only for an accurate baseline but for horsepower output to correct to common factors that make legitimate comparisons possible.

Power is all about oxygen. Fuel cannot burn unless correctly mixing with the oxygen in air. Therefore, denser air will result in more oxygen mixing with fuel for a higher percentage of burn, which creates more pressure that in turn creates more torque and thus horsepower. Less air will have the opposite result.

- Higher altitude air is thinner while atmospheric weight of the air above causes sea-level air to gain density. For example, depending on which reputable organization one listens to, the power loss is between 3% and 4% per 1000-feet increase in altitude.

- Humidity also affects oxygen content. Thus, higher oxygen density cool, dry air will generate more power than warm, moist air. Without correcting air density and humidity to a common level, comparisons are impossible.

- Beware thinking that comparison testing in the same location will negate these temperature and barometric factors. Temperature and humidity changes all day long and so do testing conditions. For example, there will be approximately 1.5 % horsepower loss for every 10-degrees F air temperature above Fahrenheit.

All modern dynos have automatic correcting computer software. They incorporate a unit, whose function is to constantly monitor barometric pressure, temperature and humidity and feed this information into the computer for corrective graphing of power output. This is what makes accurate comparisons possible no matter where the testing conducts as long as one more criterion is the

The Correction Factor Standard must be the same for Accurate Comparisons

As the reader will learn, there are many different types of horsepower for differing applications such as hydraulics, as well as for differing countries, and units of measurement. The same hold true for correction factor baselines.

While a dodgy dyno operator will have difficulty in

manipulating the automatic actual correcting barometric, temperature and humidity factors, the choice of which standard to bring them to be is easy. Different standards will give slightly but significant different horsepower figures. Please bear with me as I go through some of the various standards.

1. First, Uncorrected, which lists on the computer software, indicates that no application of correction factors for air density, barometric pressure and humidity applies to the raw data. Depending on the barometric conditions, this may leave higher or lower readings than properly corrected ones. Raw data is meaningless unless corrected.

2. The Japanese use a JIS (Japanese Industrial Standards), correction standard that will bring horsepower figures slightly higher than the SAE standard in use in North America.

3. The German DIN (Deutsches Institut für Normung), and EEC (European Economic Community, the former name of the European Community) correction standards again bring horsepower figures slightly higher than the SAE.

4. STD, which also known, as STP is a former USA/Canada standard. The Society of Automotive Engineers (SAE) set it. STD corrected figures are the highest of all mentioned here.

Therefore, there will be a temptation to correct figures to STD rather than the currently accepted SAE. The STD standard measures at 29.92 inches of (atmosphere) mercury pressure, 0% humidity at 60 degrees Fahrenheit. However, the STD standard replaces with the SAE one, in 1990.

5. SAE is the current one in use today. Ensure your dyno operator corrects to SAE. The new SAE system adopted in

1990 includes a new factor, frictional loss of horsepower. This factoring is only as accurate as the friction estimate, which will vary from bike to bike and model to model.

A chain, belt or shaft drive single or multi cylinder bikes will have different frictional loss estimates but regardless, the new system is more accurate since it at least tries to enter friction into the equation.

The new SAE formula uses an estimate of a 15% loss due to friction and increases the temperature from 60 to 77 degrees Fahrenheit and the barometric pressure decreases from 29.92 to 29.23 inches of mercury.

SAE will give a lower horsepower reading than STD or any of the other standards in use around the world.

The important thing to realize is that it does not matter what the standards are as long as everyone agrees to them.

If correction factors do not appear with the dyno graph in the ad, ignore it as unreliable. If included, ensure they are the same standards in use with comparison readings.

Mathematics can convert one standard to another. However, the dyno operator is suspect if caught using a standard other than SAE and is guilty if using different correction standards for comparison horsepower gains or losses.

## Do You Think I am Kidding?

During my research, I found a large and prominent automotive company website that states in not-so-small print

that all dyno numbers for their vehicles use a STD correction factor or they are uncorrected. There can be no other reason for this but to exaggerate their horsepower and torque claims relative to other companies correctly using the SAE correction factoring for comparisons with North American vehicles. They also must assume, quite correctly, that the vast majority of consumers will not understand what their disclosure means.

Geez, they did not stipulate if the numbers were engine or rear wheel either.

Smoothing the Dyno Graph

Whether an operator smoothes a dyno graph or not can add or subtract about one horsepower. Correct procedure is to correct to SAE and smooth the graph to five (SAE/5). A smoothing factor of zero does not smooth the graph lines at all but roughens them so that the peaks are a little higher,

thus reading slightly higher horsepower and torque numbers. Raising the smoothing factor progressively up to five will moderate the jaggedness of the graph peaks.

Ensure the dyno operator corrects to SAE/5 for accurate comparisons. The majority of dyno operators do this.

## In What Gear is Dyno Testing and Graphing Performed?

Many dyno operators like to use fourth gear, which I am sure is a throwback to the 4-speed transmission while some others like to use fifth.

Now, I am sure this will complicate with the TC96 6-speed transmission. Which, will read the highest horsepower and torque, or dyno testing? Well, there is no clear-cut answer since gear ratios and tire pressures as well as tire rating for speed will affect the torque and horsepower readings.

I will eliminate the tire rating as a factor with a Harley-Davidson since they do not go fast enough to affect the classification of the tires to any substantial degree since a minimum H speed rating allows safe sustained speed at 130mph. Many H-D tires, especially performance ones, increase the sustained-speeds-rating very much higher.

In general, I can state that testing in fifth gear will result in 2 to 5 ft-lbs of torque more at higher power band rpms than fourth gear. Sixth gear will have an increase over fifth. Therefore, it is important to know which gear testing occurs in for comparative purposes once again.

Dyno testing for comparative purposes is a minefield for the uninitiated but simple for those with a little knowledge.

The kicker is that the higher gear loses its advantage over the lower one as rpm increases above the power band. However, gear ratios and tire pressure can change this again.

Make sure, to inflate tires when cold to manufacturer specifications.

The bottom line is that when comparing products or bikes, make sure the testing occurs in the same gear, whether it be fourth, fifth, or sixth.

Air Fuel Ratios and Power

The Stoichiometric air-fuel-ratio (AFR), of 14.7:1, which I discuss at length in Chapter I: The Platform for Performance, subsection, Stoichiometric Air Fuel Tendonitis and Chapter III: H-D 110 Cubic Inch CVO, subsection, EPA Perfection:

Stoichiometric Air Fuel Mixture facilitates burning hydrocarbons, carbon monoxide and nitrogen oxides to the highest levels.

EPA approved and thus environmentally friendly 14.7:1 clean burning ratio is not where the most power produces. Many times, especially in the air-shrouded rear cylinder the perfect combusting Stoichiometric AFR will increase up to 16:1 AFR, which is injurious because of the generating of extreme heat. Paradoxically, even gas mileage suffers because of the lack of power producing from such a lean mixture even though one might logically think there is a gas saving.

The ratio is lower typically between 12.5 and 13.5:1 AFR and usually closer to 13.0:1 to 13.2:1. If enriching the AFR ratio much below 12.5:1, the result will be a too rich condition causing power loss not gain. The only possible benefit for an overly rich mixture is the cooling effect the gas will have on engine temperature. However, the bad effect

definitely outweighs the good cooling effect. This AFR range below 12.5:1 will only waste fuel as the mixture becomes too fat.

Acceleration AFR is typically best around a 13:1mixture while best cruising occurs just under Stoichiometric AFR. The goal is to try to achieve these AFR's throughout the entire rpm range at full throttle.

Many intertwining factors need to coordinate to achieve ideal power producing AFR ratios. For example, to understand the complexity, too much exhaust backpressure will negate horsepower torque increases from a freer flowing cam profile that will produce more power if the exhaust cooperated in the breathing department.

The air fuel ratio is the most accurate way to determine if

a bike is in proper tune.

## The Dyno AFR Line Graph

The air/fuel line on a dyno graph should be a straight line and every bike should have this straight line. However, the line will probably be wavy on a carbureted bike without enough backpressure. If dyno tuning your bike and the air fuel line is missing, it is because the dyno operator is hiding it.

The goal is a straight line on every bike and thus every dyno graph. On the other hand, it is normal for the HP and torque curves to differ on every bike. The relatively new air-fuel sniffer is very necessary on modern dynos to help analyse positives and negatives. It is important to utilize this add on older dynos as an upgrade.

Carbureted bikes have more difficulty in achieving a

straight air/fuel line. Unscrupulous, lazy or lack of skill operators may hide this line when dynoing a carbureted motorcycle because it involves too much work in dealing with customer queries or straightening the line.

Time is money and these guys want the money without the hassle.

Carburetor Tuning Conundrums

However, the tuner does have a real problem in that fixing an aberrant air-fuel line in one area may make it worse in another because carburetor jetting and needle adjustment and tuning overlaps into other rpm ranges.

- The main jet controls initial full throttle from that rpm up the full range. Changing the carburetor main jet bigger or

smaller will move the entire full throttle curve up or down. For example, the leaning out a rich spot at a lower rpm will further lean out an already lean area at a higher rpm.

- Furthermore, if looking at an opposite scenario, enriching a high rpm lean spot will also further enrich an already rich condition in the midrange, adversely affecting torque lower down in the rpm band. Mixture disparities like the simple example of midrange versus high range at full throttle are part of life with a carburetor.

Hence, another advantage of the more precise electronic fuel injection.

- There are other carburetor mixture conundrums with the other fuel circuits. Cams and other performance work can further complicate the fuel mixture challenges.

The dyno operator knows, as does any competent mechanic that carburetion has overlapping circuits where what is good at one rpm and throttle setting will in all likelihood negatively impact at another rpm. Therefore, any jetting or needle changes are necessarily compromise ones. This overlapping makes a straight air fuel line onerous, time consuming and in all likelihood may not be possible to achieve.

## The Exhaust System to the Rescue

However, exhaust changes will help immensely in correcting rich conditions because the backpressure pulse waves can change the ease or difficulty of scavenging burned exhaust gases. Therefore, as a generalization I can say that:

- An overly free flowing exhaust will usually cause a mid range dip of the air fuel curve, which is the same as saying the

AFR ratio enriches too much below an acceptable lower range of 12:1. This enrichment increase translates into a drop in torque ft-lbs of power in this rpm range.

- This situation exacerbates with performance cams whereby the torque drop increases without an exhaust change. Correcting this condition through carburetor jetting alone will certainly lean out the engine higher up to the point the increasing leanness (higher AFR ratio) as the engine progresses into higher ranges may even damage the engine. The balancing act continues.

- The exhaust system change will do much to correct this AFR dip with carburetor tuning coming afterwards to finesse the AFR line straightening.

This is why the air sniffer component of the dynamometer is so important. The operator skill set demands increase.

This work requires talent, knowledge and practical experience in that the AFR line never achieves perfection with carburetion. Carburetion air fuel lines that stay between 12 and 14:1 AFR are about as good as good can get. Now this is not a big deal but an acceptable reality when educated.

What Should a Dyno Operator Look For?

1. It is obvious once said. Air, fuel, exhaust and ignition are the main areas of interest. Carburetor jetting, EFI fuel modification, ignition timing, sufficient air intake and sufficient air outtake also known as exhaust backpressure all interweave to increase or decrease power substantially.

2. Tuning each in isolation can afford a few extra ponies but the next tuning operation may take away from the first.

Tuning must encompass all these factors in combination with each other not in isolation.

3. Different problems require differing techniques or order of tuning. Ensure the restrictive emissions control air filter and air box replace with freer flowing ones. There are many great ones on the market, which I examine in Chapter IX: Fuel subsection, Air Cleaner.

4. I favor installing a correct exhaust next since the engine is much like the human lung where air must breathe in and breathe away without impediment.

5. Do not follow my thoughts religiously because differing situations may rearrange the priority list I am inadvertently developing here.

With hemi-head Harley-Davidson's like the Shovel, Pan, Knuckle and Ironhead XL, exhaust tuning moves up the priority scale, as this is where the major problems first occur. Evolutions may use timing further up the priority list whereas in Twin Cams it may not be as important.

The dyno is wonderful in that this tool will immediately indicate if changes are in the right direction or not.

6. There may be some more exhaust tuning afterwards.

7. Now I look at fuel management whether it is through carburetor jetting or the more precise EFI remapping.

8. Finally, ignition timing may be the last significant area I look at, which is not to denigrate its importance.

Evolution timing is adjustable. I can move it back and

forth while on the dyno, which indicates the right direction to move and finally the ideal timing setting.

Twin Cams are non-adjustable through operator finessing. Ignition changes require a fuel modifier with an ignition component. Input modifiers that change sensor signals before entering the ignition control module (ECM), also adjust ignition timing since the fuel tables' tie into the ignition tables also known as the map or matrix mapping. Most output modifiers that install after the ECM only modify the fuel map. Power Commander is an output-modifier exception because it also has a separate ignition function.

- Carburetor jetting is predictable if tuning a stock Harley or one that has freer flowing intake and exhaust breathing. Predictability is out the window with further performance changes and tuning must be on a case-by-case basis.

- Stock Evos and more so with ever EPA restrictive Twin Cams need freer breathing for both improved engine efficiency and further performance gains.

- I should note that jetting changes do not necessarily follow with better breathing. I realize this sounds illogical but Harley-Davidson, if meeting EPA dictates below minimum jetting or ECM settings will raise the AFR to just below or meeting the regulations.

The short answer is that bikes will often arrive from the Factory with a slightly rich AFR. This means that sometimes the engine breathing changes can grow into the Factory settings without further change. Let me make one thing abundantly clear. Never assume this and always check for an acceptable AFR ratio, which means the same as correct jetting or mapping as the case may be.

# Timing/Carburetor Jetting Recommendations

These recommendations are for stock models or those with freer flowing intake and exhaust. They are starting points for correct jetting but always check the air-fuel-ratio to confirm correct jetting. Jetting is for the main jet only. Timing is for an ideal starting point. Surprisingly stock timing or close to it will work well.

Timing and Ignition Table

Note: TC96 Twin Cams are fuel injected.

Drag pipes cause midrange rpm's to become too rich, which lowers torque; carburetor jetting cannot effectively compensate for this because other areas are negatively affected with midrange leaning of rich AFR, only more backpressure can effectively correct rich condition. An example, I will repeat is that an astute technician-operator will address exhaust backpressure before tuning a carburetor because backpressure increases or decreases will have a major affect on a problem AFR at a specific rpm range but not affect another that is fine.

Carburetors have overlapping jets and needles that are best left for fine-tuning after other major changes.

If plotting an air-fuel-graph-line on a chart relative to rpm, it is desirable to have the line as straight as possible for an efficiently running motorcycle. A straight graph line indicates that the air-fuel-ratio remains correct over the rpm range. If it becomes disjointed, it means that the air-fuel-ratio is skewing too rich or too lean.

It is easier for EFI bikes to get this AFR graph line straight. If the AFR line is not straight or made not to be straight by unprincipled dyno operators with dollar signs in their eyes, a fix may or may not be necessary depending on the cause of AFR line aberration. The fix is an output fuel or an input fuel-ignition modifier as an ECM tuning aid. I love these modifiers

and use them repeatedly on bike after bike.

However, dishonest tuners can set the stage for their sale and installation labor to resolve a perceptual problem that maybe was not a problem at all. Fortunately, the excellent modifiers on the market like Power Commander (www.powercommander.com), Terminal Velocity (www.terrycomponents.com), Thunder Max (www.thunder-max.com) and carbureted S&S Intelligent Spark Technology (IST) (www.sscycle.com), and many others will all give superior AFR ratios, eliminate flat spots, hiccups and develop more power.

I feel it important to install them for the right reasons. I always advise riders to find a competent mechanic that is trustworthy. I say the same regarding the choice of a dyno operator and there are many excellent ones, some examples

are Dr Dyno aka Fred Eno (www.drdyno.com), or Speed's Performance Plus (www.speedsperformanceplus.com), and R&R Cycles (www.rrcycles.biz, to name a few.

## A Superior Tuning Aid Will Lead to Better-quality Tuning

At least this is the theory. However, the dyno operator needs the skill and the morality to utilize fully this wonderful tool to the customer's advantage.

The dynamometer like a flow bench is a superlative tool for research and development, tuning and isolating power-robbing situations.

## Will a Dyno Hurt an Engine or Power Train?

Can a dyno hurt a bike engine or its powertrain? The answer is yes and no. Again, the difference is not with the

dyno but with the operator.

I do not like putting a new engine on a dyno until there are some miles on it. This is a personal preference, as I must note a dyno can have use as a break-in tool. I certainly will not put a bike on a dyno unless there is a blower pouring fresh relatively cool air onto the engine. Remember that a Harley-Davidson is air-cooled.

I am developing a list of what to look for before allowing a bike onto a dyno. Much the same list appears earlier in what to look for in a dyno operator and equipment. The following list is from a safety perspective for bike longevity. Dynos or more properly stated dyno operators can cause premature wear and breakage to a motorcycle and its component parts if not applying the dynomometer correctly.

1. Do not blindly trust every operator. Also, do not assume everyone is unscrupulous because it is not true nor fair. In addition, a guy like me will not work on your bike if I feel customer mistrust. Walk a fine line politely.

2. Check out the dyno and the operator first by simply watching and observing.

3. If the operator revs the bikes on the dyno beyond their power bands, walk away.

4. Once is okay, just past the power band because the operator needs this information of how far to rev the bike for testing and tuning.

5. If a bike is cold, ensure the operator respectfully allows the dyno to warm it up before testing the power band limits.

6. What is the use of a dyno, if the operator does not have tuning tools?

7. Again, there must be a source of cooling air. If there is none, engine damage may ensue.

8. It is easy to determine if an operator respects the bike on his dyno. A good and positive sign is if the dyno operator is treating the bikes with care.

It is true that a properly operating dyno and a cautious, caring technician can still snap a chain, break a belt or cause engine break down. Do not blame this on the dyno or the operator, as whatever happens would happen very soon on the road anyway. In other words, the chain or belt or engine component was about to break anyway because of improper maintenance on the owner's part.

# Some Different Types of Engine Horsepower

There are many ways of expressing and measuring horsepower although some are now dated. Most are now out of use although they still hang on with older people and for certain applications. I am saving the good stuff for last while I get the ineffective and virtually useless definitions and measuring techniques out of the way. I only mention these, so the reader will recognize what they are about if reading these types of horsepower in print.

## Nominal Horsepower

Nominal Horsepower (nhp) is inaccurate. Nhp is better to view as an educated but guesstimated horsepower. This guessing horsepower is out of date, because it applies to steam engines, which is just as well because of its inherent

inaccuracy.

The calculation of nhp involves engine size and piston speed. It only approaches accuracy at a cylinder pressure of 7 psi with piston speed between 180 to 248 feet per second (ft/s).

## Gross Horsepower

Gross Horsepower is the same as Indicated Horsepower (ihp). Do not confuse Gross Horsepower with SAE Gross Horsepower, which is different. Indicated horsepower is the theoretical power of a reciprocating engine if it is completely efficient in converting the energy contained in the expanding gases in the cylinders.

Ihp's theoretical capability of an engine ignores frictional

losses within the engine such as bearing drag, rod and crankshaft windage losses, and oil drag, etc. Gross (ihp) horsepower is misleading because the mechanical efficiency of an engine means that the actual power output may only be 70% to 90% of the indicated horsepower. It calculates from the pressures developing in the cylinders that measure by a device called an engine hence, the name indicated horsepower. Ihp was in use for measuring steam power in the 19th century.

SAE Gross Horsepower

Prior to the 1971, American automakers rated their engines in terms of SAE gross horsepower under SAE standards J245 and J1995. Gross Horsepower measures using a blueprinted test engine running on a tooling fixture, minus air cleaner, mufflers, or emissions control equipment but sometimes with testing exhaust headers instead of stock ones.

The atmospheric correction standards that include barometric pressure, humidity and temperature utilized in obtaining the Gross Horsepower ratings was unrealistic by being overly positive.

Gross Horsepower and torque figures reflected a maximum, theoretical value that in actuality overstated the power when the engine became part of the powertrain. Complicating matters and leading to consumer mistrust was that manufacturers reportedly adjusted horsepower figures for marketing purposes. Therefore, power ratings of mass-market engines had exaggerated horsepower claims.

Conversely, muscle cars of the era had the reputation of understating their power claims. In fact, the pre-1972 muscle cars were more accurate in reporting actual horsepower with

most thinking them understated. In truth, testing indicates that they rarely if ever produced an advertised Gross Horsepower never mind exceeding them. The reality is that muscle cars were closer to reporting actual Gross Horsepower than family cars.

## Shaft Horsepower

Shaft Horsepower measures and describes power at the output shaft of the transmission, minus frictional losses. We do not need to understand anymore about Shaft Horsepower since it will never be applicable to anything of worth on a Harley-Davidson. I am not sure where it is of value anywhere else either unless the transmission hooks up to machinery but then we are not talking about a vehicle.

## Brake Horsepower

This is where the good stuff starts. Brake Horsepower (bhp) is the same as Net/Crankshaft Horsepower, which is a much more descriptive term. This is engine horsepower measuring at the flywheel sprocket shaft before it interacts and delivers torque to the powertrain. Therefore, Brake Horsepower does not account for frictional and dissipating losses in the powertrain. It also does not take into account any auxiliary components such as a generator or alternator, or speedometer drive.

Most manufacturers' like Harley-Davidson use Brake Horsepower when advertising their products because it is always higher than Rear Wheel Horsepower. The name is a little confusing. The brake prefix to horsepower refers to where the power measures which, of course is as I said at the engine's output sprocket shaft. Well, that tidbit of information did not help resolve the confusion, so let us pursue this a little further. Brake Horsepower measures on an engine

dynamometer not to be confused with a chassis or rear wheel dyno. The term brake refers to the original use of a dynamometer band brake that measures torque during the dyno run.

As we know, torque converts to horsepower by a mathematical calculation in which rpm divides by the constant 5252 multiplied by the torque.

Rear Wheel Horsepower

Rear Wheel Horsepower (whp) is the same as True Horsepower (thp), or Effective Horsepower. Rear Wheel Horsepower is Brake Horsepower minus the power losses deriving from powertrain torque transfer and parasitic auxiliary components. Rear Wheel Horsepower is my favourite way of describing and comparing power in a motorcycle because it is real power delivered to the rear-tire-contact-

patch where it meets the pavement.

The Society of Automotive Engineers began to discourage the terminology Brake Horsepower (bhp), in favor of horsepower (hp or HP), to describe the net power of the engine at the sprocket output shaft.

## SAE Net Horsepower

SAE Net Horsepower began use in 1972 under SAE Standard J1349. This reflects the rated power of the engine in as-installed trim, with all accessories and standard intake and exhaust systems. American car manufacturers began to use SAE Net Horsepower in order deflate and thus understate power ratings to allay the insurance industry as well as the environmental and safety people.

Rear Wheel Horsepower began to suffer additionally with the addition of emissions control components, so the manufacturers could hide these further losses to the consumer by using SAE Net Horsepower. Therefore real power loss is hides from the consumer while power gain diminishes for insurance, emissions and safety lobbies.

## The Worldwide Diversity of Horsepower

The concept of a measurement of work evolved simultaneously and many times independently all over the world. The usable definition describing work performance and a quantifiable measuring concept that allows comparisons and rankings of different vehicles and machinery is radically different from one country or continent to another. To add to the mix are refinements to the base definition to accommodate specific types of comparisons and rankings.

If the reader is from North America, he or she will undoubtedly be familiar with and use Watt horsepower.

## Watt Horsepower

As discussed earlier James Watt arbitrarily decided upon 33,000 ft-lbs of work per minute would define one horsepower. The two components to his decision are distance and weight. If a horse pulls 3300 pounds a distance of ten feet in one minute, this constitutes one horsepower (10 feet x 3300 pounds = 33,000 ft-lbs). Similarly, if the horse pulls 1650 pounds for twenty feet in one minute, this will also constitute one horsepower (20 feet x 1650 pounds = 33,000 ft-lbs). Therefore, less weight pulled a greater distance than a greater weight pulled a lesser distance will always equal one horsepower if the multiple of distance and weight equals 33,000 ft-lbs.

Perhaps, James Watt is most famous today for the wattage of a light bulb.

Watt`s capricious defining of a horsepower being capable of producing 33,000 ft-lbs of work in one minute translates into many other units of measurement the reader is surely aware of but unaware that a horsepower defines them. Let us look at the watts in a light bulb first since it has an obvious connection to the horsepower guru Watt. The mythical Watt horse can generate 746 watts of continuous light by expending one horsepower of energy on a treadmill! Therefore, one-Watt horsepower equals 746 watts. Watts are interrelated with volts and amperes (AMPS), as well.

British Thermal Units (BTU), a measurement of heat also derives from one horsepower. A BTU is the amount of energy

required to raise the temperature of one pound of water one degree Fahrenheit. One horsepower equals 746 watts equal 2,542 BTU's. Therefore, one horsepower or 746 watts can generate 2545 BTU's or put another way will raise 2545 pounds of water, 1 in one hour.

Again, the temperature and the amount of water weight can vary with each other as long as the multiple of both equals 2542. Next is the joule, which represents by the symbol J. A joule is the mechanical equivalent of heat. Therefore a joule is the number of units of work in which a unit of heat can perform.

James Joule found its value in experiments that showed the mechanical energy Joule's equivalent. One BTU is equal to 1,055 joules. This keeps going as one-joule equals or 60.24 gram calories or .06 food calories. On it goes, all from an arbitrary figure plucked out of the air to describe a measurement for work. To tie this into a circle, a horse

producing one horsepower would burn about 640 calories in one hour.

However, at the end of it all, science does not really like the term horsepower and what flows from it because of its arbitrary and imprecise beginnings.

## Mechanical Horsepower

Mechanical horsepower is another word for a Watt Horsepower where one horsepower equals 33,000 ft-lbs per minute, 550 ft-lbs per second or 745.699 watts, which mercifully rounds out to 746 watts.

## Metric Horsepower

Metric Horsepower is the metric equivalent of a Watt horsepower or mechanical horsepower and converts to 75 kgf.m/s, which in long form is 75 kilogram-force meters per second. However, the conversion is also imprecise since 75 kgf.m/s equals exactly 735.499 watts and not the expected 746 watts. Therefore, a Metric Horsepower really is only about 98.6% of mechanical horsepower. The conspiracy theorists out there will immediately understand that some vehicle manufacturers may fudge their horsepower figures upwards by 1.4% because of this inequity.

The various short form replacements for HP (horsepower), when using metric horsepower depending on country and language are ch (French cheveaux or horses), cv (French Pferdestarke), CV (Italian Cavalli, Spanish Caballos and Portuguese Cavalos), hk (Swedish hastkraft and the Norwegian and Danish hestekraft), hv (Finnish hevosvoima), LE (Hungarian loero), pk (Dutch paardenkracht), and finally PS (German Pferdestarke or horse strength).

Europe's EEC (European Economic Community) made obsolete the usage of these terminologies in 1992 in favor of the kilowatt. This is now the official member-country measurement unit for (horse) power. The metric kilowatt is 1000 watts.

However, the old terms are still freely in use because older people, in particular are familiar with them and not the kilowatt.

They all translate into 98.6% of mechanical horsepower.

Electrical Horsepower

Electrical Horsepower converts to 745.699 (746), watts.

This type of power measurement has use in the electrical industry for electrical machine power output. It defines exactly at 746 watts at 100% efficiency.

However, it is next-to-impossible for electric motors to run at 100% efficiency. Electric motors usually have a specification plate attachment on them. It is important to know that the specification listing is for their motor power output and not their power input capability.

Boiler Horsepower

Boiler Horsepower converts to 33,475 BTU per hour, which is the energy rate needed to evaporate 34.5 lb (15.65 kg) of water at F or the equivalent Celsius in one hour. This form of horsepower has use in boiler rooms and power plants.

Hydraulic Horsepower

Hydraulic Horsepower is the same as mechanical or watt horsepower but expresses as a flow rate for fluid, U.S. gallons per minute (US gal/m). It can also derive from the formula of pressure ÷ which I will not go into here as I am getting too far off topic.

RAC Horsepower

This one is a beauty! RAC Horsepower relates expressly to a piston engine like a Harley-Davidson. One liter of displacement equals 10-RAC horsepower. The metric liter converts to 61.01 cubic inches.

RAC also known colloquially as tax horsepower is the acronym for Royal Automobile Club. RAC Horsepower was in use to levy a road tax. Taxation for license plate purchase or renewal was based on taxable horsepower.

It’s use was not confined solely to European countries like Great Britain, Germany and France as well as others but to some American states such as Illinois.

The determination of taxable horsepower was by a simple math formula basing on cylinder swept volume, which measures in liters or the equivalent cubic inches (c.i), or cubic centimeters (cc).

It gets weirder. The British RAC Horsepower rating only related to total piston area. Therefore, the Brits seeking to lower tax ratings began developing cars and bikes with very long strokes and narrow cylinder bores.

Therefore, if the reader is interested in older British cars or motorcycles, you will find these seemingly perplexing long

strokes relative to cylinder bore diameters.

## Drawbar Horsepower

Drawbar Horsepower uses the acronym dbhp. This is the horsepower a train locomotive has available to pull its rail cars. Drawbar Horsepower also measures a farm tractor`s ability to pull agricultural machinery.

There is no calculation formula to determine drawbar horsepower. Drawbar Horsepower measures by a special dynamometer rail car attaching to the locomotive that records speed and the pull exerted by the locomotive. Maximum power available measures with a second locomotive with brakes applied.

Then the power capability of the locomotive to pull a

certain number of rail cars with their respective loads can be calculated with the pull and speed figures. I do not think I need to explain any further since this book is about a Twin Cam motorcycle.

## Matching Horsepower and Torque to Rider Needs and Wants

Rpm, horsepower and torque relate directly, in fact, mathematically so.

The rider must match horsepower and torque to their particular need. If the rider is into stoplight-to-stoplight racing, then torque is more important than rpm, torque multiplied by rpm equals horsepower.

As a generalization, increasing torque relative to rpm will lower the Powerband whereas increasing rpm relative to torque will raise the Powerband. If the rider is, primarily into

highway, touring then horsepower derived from less torque and more rpm will positively influence this style riding.

## Conclusion

To think the whimsical concept of horsepower all started with some guy trying to figure out how much coal a pony could pull up a shaft in a certain time-period. Furthermore, the bounds of credibility stretch unbelievably thin, when the same guy multiplies his crude calculations by 50% because his test pony was smaller than a horse.

It is mind blowing that James Watt's arbitrary calculations are the measure of so many forms of power today including light bulb wattage. So imbued are his definitions and calculations inter-relating differing forms of power that they are here to stay.

Most people talk in terms of horsepower but because it is such a capricious concept, perhaps torque, which is a real world meaningful measurement of twisting force should be the terminology of choice.

After all, horsepower is but a mathematical concept.

Torque is real; it accelerates your Harley-Davidson.

# Chapter VIII

## Eliminating the Performance Obstructions

Originally, my plan was to deal solely with insufficient head flow in this chapter, the greatest barrier to more power with both the Twin Cam 88 and the TC96. However, I am going to broaden this subject because there are a series of obstructions to performancing including the heads, of course, but also the cams, ignition, fuel delivery, and the exhaust system, in short the restrictive emissions controls. It is not that these components are of bad design or are substandard. They include the newest technologies ever envisaged on a Harley-Davidson with the exception of the V-Rod.

The problem is that the design of these components that all link together and support each other has tunnel vision. Their goal is to eliminate harmful emissions to the environment. Unfortunately, other considerations like power and more comfort from less heat are secondary and subservient.

Harley-Davidson is in a difficult situation. The question for the Motor Company becomes how to comply with the EPA and CARB while keeping a tradition alive.

The tradition we all want to protect and help endure is dinosaur technology involving a valve train versus overhead cams; an awkward cylinder angle with equally uncomfortable ignition firing order; inline air-cooled cylinders and only two-cylinders when more would form a symbiotic relationship assisting each other. There is nothing here that would not immediately resolve with overhead cams; liquid cooling with

radiator; a wider cylinder angularity, and effective ignition firing order with more cylinders.

The rock is tradition and the hard place is complying with emissions controls.

See Chapter IV: Cams, Cams, Cams; Chapter IX: Fuel Delivery that addresses the cams, ignition and fuel delivery in much detail. This chapter will deal with the head obstructions and overly restrictive exhaust systems.

## Key to Power: Head Flow

Air is power!

Fuel must mix proportionately with air at about 14.5-parts air to one part fuel for cruising and hot idle. For acceleration and cold idle an air-fuel-ratio (AFR) of around 13:1 give or

take a few points of 13-parts air to one part fuel. If there is insufficient air to mix with the fuel in these generalized proportions then combustion and power will suffer.

It is easy to jet or squirt more fuel into the cylinder. The process of fuel delivery is not problematic.

The delivery of air, on the other hand, requires an unrestrictive air filter including an air horn, a complementary intake induction tract, properly ported manifold and head ports, with sufficient valve sizing along with an unrestrictive exhaust with backpressure.

Once, the above conditions are in place, the cams become the masters of flow allowing many laws of physics to do their thing.

Add complementary ignition timing and some compression and the engine will deliver power.

## Capitalism at its Finest

No one knew anything about the Twin Cam in 1998-1999 except for the Factory people involved in its development and production plus the lucky few like myself. Select management kept the secret so well that dealers knew nothing nor did most other Factory employees and management.

In fact, dealers became part of the rumor mill, repeating bad information, half-truths, and no-truths. Everyone knew that something extraordinary was coming down the pipe. Then, the introduction of the Twin Cam caught everyone by surprise, utilizing design and technology never seen before on a traditional Harley-Davidson.

This necessary secrecy from competitors and especially the powerful aftermarket created a huge information void. Many of the new business-type dealers do not always have a full mechanical knowledge. Dealers were scrambling to answer all the questions put to them by an enthusiastic and demanding customer base.

Up Selling

However, the dealers do know how to up-sell and sell they did.

A strange thing began to occur soon after the introduction of the Twin Cam in 1999. Without regard or knowledge for the limited-flow capabilities of the heads, most dealers began pushing the 1550cc (95 cu in), upgrade kit for the new engine. Who was to know that the exquisite head design and

metallurgy had poor flow characteristics for such a big engine? In addition, who would have suspected that the venerable Harley-Davidson would begin to evolve this type of engine design with a view to marketing upgraded performance parts?

In my opinion, it not fair to the consumer if some parts are under-engineered for a purpose of creating an aftermarket, instead of designing and incorporating these improvements into the finished product. However, this may simply be another case of the Motor Company having to meet emissions controls to the detriment of potential reliable performance.

Nevertheless, to be realistic, this is capitalism at its finest.

## Disappointing Power Expectations

I can unashamedly say that I was taken in when first

observing the Twin Cam head design in Milwaukee in early 1998 because it was obviously superior at first glance. Their design is superb. The metallurgy is the best. Even the powdercoating, whether it is black wrinkle or blond is bulletproof.

However, in my opinion, the original TC88 heads are deficient in the flow of air.

The newer TC96 heads are better but, in my opinion, also deficient for the size engine they serve.

The TC88/TC96 bathtubbed combustion chamber shape I was familiar with on high-performance aftermarket Evo heads led me to draw wrong conclusions as to head flow. The heads look so good that I made a misassumption as to their flow capabilities.I was not alone.

Squish Bands

The flattop round piston ascends in its cylinder and squishes up against the bathtub design. Squish forces the air/gas mixture turbulently into the chamber, mixing air and gas better than any previous H-D head design. The proportionately mixed air and gasoline molecules induce superior flame travel and a much higher percentage of burn.

The TC88 compression ratio is higher, 8.8 to 9:1, (TC96 is 9.2:1) compared with the Evolution's 8.5:1. This increased compression leads to more torque and thus horsepower.

So why do other dealers keep selling 95" big-bore kits with no headwork to accommodate the increased cubic inches? Why do they not educate the consumer that sufficient air delivery is essential to the bigger cubic-inch displacement in

order for the increased displacement to work to its true potential? Ignorance of the product is the only thing that I can think of.

In fairness, everyone was on a very steep learning curve with the introduction of the Twin Cam. The introduction was such a well-kept secret.

The reason is partially due to H-D creating its own aftermarket, for the first time, in necessary secrecy, well in advance of everyone else in the market place.

## The Heads

Bathtub-style combustion chambers are the first thing noticeable when the engine is apart.

When assembled, the fat finning for more cooling is most striking with a 60-percent increase over the Evolution. With the bathtubbing, there is less room between the valve seat insert and the spark plug hole.

Therefore, reducing the traditional Big Twin 14mm spark plug threads to 12mm became necessary. If left at 14mm, the engineers felt that it might cause a weakness.

The 242 alloys are the same used in the Evo heads.

The bathtubbing is more a rounded rectangular shape than the bathtub or kidney shaped configuration that is prevalent in the aftermarket. Previous hemi-heads found on the Shovel, Ironhead XL, Pans, and Knuckleheads have rounded hemispherical inverted combustion chambers. The piston top or dome is also rounded and upturned in order to fit into the

hemi-head. This system is not very good for uninterrupted flame travel or a high percentage of burn because of the interfering piston dome and lack of squish.

The Evos have a D-shape combustion chamber. Inside the D, the chamber cuts off instead of continuing in a perfect circle. The rest of the circle is flat and even with the surrounding gasket surface. A flat top piston squeezes up against this flat squish squishing the air/gas mixture turbulently into the smaller D chamber promoting better flame travel and a much higher percentage of burn.

The TC88 bathtub chamber is small, having only 85cc volume. With bathtubbing there is a flat squish band all around the chamber so that the big 3 ¾" bore flat-topped piston squishes the air/gas even more turbulently into the chamber, promoting the best flame travel and highest

percentage of burn of the different historical head configurations.

The permanent mold head has the same size intake valve as the Evo, but the exhaust is about .030” smaller in diameter but a little thicker. An indent mark in its center face allows for easy recognition.

The two length pushrods are also different colors for the same reason.

Pushrod angle holes are almost parallel straight up and down. Evo and earlier motor pushrod angularity with their resultant problems eliminates with dual cams.

Combining with this smaller exhaust valve is a smaller diameter, reworked exhaust port that takes advantage of Bernoulli’s principle about getting the exhaust gasses exiting

faster and more efficiently to help with cooling and emissions considerations. The built-in torque valve is more correctly an anti-reversion (AR), valve. The AR is not really a traditional valve but a step in the exhaust port to help confused gases exit the exhaust pipe.

The TC96 eliminates the anti reversion step from the external mouth of its exhaust port.

Anti-Reversion

Removal of the anti-reversion step at the outside of the TC96 exhaust port opens up the port considerably allowing much more flow.

This anti-reversion step was a design improvement change in 1999 with the advent of the Twin Cam 88 even though it

restricts the all important exhaust flow properties of the head design. Exiting gases partially overcome this by speeding up since the same amount of gas must pass in the same time according to a physics principle. Anti-reversion is also extremely important as helps prevent exiting exhaust gases from returning and soiling the fresh intake charge in the combustion chamber. As exhaust-pulses force out through the exhaust valve, out the port and into the header system to the mufflers, Kinetic wave exhaust-gas pulses will return up the pipe many times before finally expelling out the muffler. It is of utmost importance that these bastardizing pulses of burned gases bounce off a closed exhaust valve. These reversion gases also bounce back off anti reversion steps built into the heads.

Installation of anti-reversion torque cones in the exhaust header pipes where they meet the exhaust port on the 2007 models is easily done and inexpensive. The reader can now understand why the oft-ignored exhaust system is complex as only one of its functions is to control the timing of these

power-robbing pulses.

## Porting and Polishing

The Twin Cam lungs cannot operate at peak efficiency without increasing breathing capabilities. Porting and polishing stock heads with a little valve work, such as installing the .030” larger Evolution exhaust valves will work wonders for engine efficiency and releasing suppressed torque and horsepower. The Evo valves fit easily into 1999 to 2003 heads and only require the valve seat re-cutting.

Cutting valve seats where they interface with the valves is extremely important, both for sealing the combustion chamber but also for air flow. Increase in airflow dramatically improves with quality work within one-half inch of the valve seat. This is where the majority of flow improvement will

come from.

Compare this with only a potential 1% improvement by porting and polishing the inside of the intake manifold to a lustrous shine.

Porting

Porting is reshaping the intake and exhaust tracts to encourage more flow. Valve guides that stick down into the port also need addressing. I do this in a variety of ways, such as beveling the guide itself so that a pointed as opposed to a rounded edge meets the flowing air. In some situations, a rounded guide works well also. Shaped similarly is the aluminum port material as it approaches the guide.

The reader can see that this is all about easing impediments to the flow of air. This is not work for amateurs.

It is also not work for many mechanics, either.

## The TC88 Valve Seat

One of the principle areas to address when porting TC88 heads is a severe overhang of the valve seat relative to the head port.

Valve seats press fit into the alloy heads. A very hard steel alloy withstands returning slam from the valves. For an optimal flow of air, the valve seat should radius and blend into the port it serves.

Much improved on the TC96 models is valve seat matching to the head ports from the 1999-2006 TC88 Twin Cams. While there is better matching of the seat there is regrettably still some overhang of the seat. This needs radius shaping for

better flow. Again, I cannot over-emphasize this point as most flow improvement in any porting and polishing operation occurs within one-half inch of the seat area.

## Polishing

Many misunderstand the polishing of ports. I polish exhaust ports because I do not care about the exhaust gas-mixture composition flowing out. It is waste and good riddance. Therefore, I polish the exhaust port tract to speed up the exit of bastardized gases.

It need not be a mirror shine as this is primarily for marketing purposes. Mirror polishing is not necessary for expedient elimination of purging exhaust gases. If having stock heads ported and polished some shops will polish the exhaust tract to a mirror shine. This is purely feel good stuff.

It is beautiful. However, the lustrous mirror shine is not necessary. A polished smooth surface will suffice to encourage the faster movement of the exhaust gases out of the head and into the exhaust header pipes.

The intake tract is a different story. I do not polish this port, as all our attention must direct at the porting. I leave the intake tract unpolished especially in the inefficient hemi-heads like the Shovel or the Ironhead XL to encourage the entering air/gas mixture to mix.

Theoretically, the rough texture of the port walls will cause turbulence to assist in the mixing of the fuel with air so that it is better able to burn. I will take turbulence where I can get it in older hemi-heads, as it will assist in the fuel burning.

There has always been controversy over how true this is.

There are even misguided porters who believe in polishing the hemi-head intake, but they are a small minority.

I think where the confusion may lie is in the type of head design and fuel delivery. Hemi-heads do not mix the air and fuel mixture well so they need all the help they can get in the intake tract. Therefore, turbulence in the intake tract assists in mixing the air and gas. However, turbulent air is a slower moving air. Evo and certainly Twin Cam heads with squish bands mix the air/gas efficiently so we want to get as much volume into the combustion chambers as possible.

The way to move more volume of air and gas is through laminar or smooth flow. I do not want turbulence in the intake tract in order to deliver a higher volume of air and gas into the combustion chamber.

Air cleaners perform a valuable function here, as the air travelling through the filter is turbulent and disorganized. The air horn on premium air filter backing plates like Doherty Power PACC (www.dohertymachine.com), and S&S Cycle, Inc. Super E and G (www.sscycle.com), to name two takes the slower-moving turbulent filtered air and organizes it into higher velocity laminar flow.

## Intake Port Smoothing

Delphi electronic fuel injection squirts gas towards the back of the intake valve. A carburetor begins mixing gas with the air in the venturi before the intake valve. Therefore, a carbureted hemi-head will want an unpolished intake tract. A carbureted Evo and especially a Twin Cam will want a slightly smoother intake tract. An EFI Twin Cam will want the smoothest intake tract of the examples given. The name of the game with modern combustion chambers is to increase the volume of air not slow it down.

Aftermarket atomizers designed to mix air and gas in the intake side can be very counterproductive especially in EFI equipped motorcycles with squish-combustion chambers. These atomizers come in various styles but a common one is a fine mesh screen placed at the start of the carburetor or EFI throat. They will take the air that the air cleaner design has calmed down into laminar flow and then turbulate it, which will slow its travel and therefore volume.

## Improved TC96 Head Flow

An improvement to the reworked TC96 head castings over their TC88 counterparts is the elimination of a restrictive-to-flow curvature on the inside of the turning exhaust port. This accomplishes with a welcome over-sizing of the whole port but still radiuses narrower down to the almost same size valve

seat. This narrowing down will increase flow as explained by physics, namely Bernoulli's Principle. A fluid or gas moving from a larger volume space to a smaller one must speed up. The same amount of fluid or gas will move through the narrowing in the same time-period because of the relentless pressure from behind. This pressure difference will result in a net force, which by Newton's second law will cause an acceleration of the fluid.

Factory representatives allude to the reworking or massaging of the TC96 heads. However, the casting looks to be of better quality. It is nicer with smoother edges and more defined such as oil cavities etc. although these aesthetic improvements do not affect anything mechanically; it is just visually better.

The exhaust valve (18085-05) is also more conducive to improved flow as it approaches a tapered tulip shape where the stem joins the top of the valve as opposed to the

1999-2005 more abrupt flatter shaped exhaust valve top.

At Heavy Duty Cycles, we measured some 1999-2006 earlier exhaust head ports and found an average approximate diameter is 1.330”, which varies from head to head. The new head exhaust port diameter is about 1.510 inches. This increase in sizing is huge as thousandths-of-an-inch is equivalent to miles inside an engine.

Intake ports are pretty much the same size but a noticeable and superior difference is that the seat edges and casting they fit into mate correctly on the heads I removed from my 2007 FLHXI. The improved intake port shape over previous Twin Cams allows for enhanced intake airflow. The port curvature is gentler and not as abrupt when changing course allowing smoother flow. The intake valve’s (18074-05), head is also approaching a higher flow tulip shape. The valve guide

(16465-06) has a shorter reach into the head, which is less obstructive to increased flow.

Intake manifold flanges (26993-06) are new for better angling of the manifold aperture relative to the port it serves. This is an obvious improvement.

In keeping with this, the actual flange mounting holes in the head itself are now equidistant from each other as well as the port-center. Earlier heads use front and rear flanges (27009-86A and 27010-86A) that mount to holes in the head that are offset and not equidistant. This improvement combining with the new injector angle of from the previous injector serves to help increase flow of air and fuel delivery.

The valve seal improves once again from the 2002 change, which was also superior to what went before. The seal

incorporates into the lower valve collar since the 2002 change. It appears to incorporate an improved more pliant rubber-like material that will hopefully have a longer service life. The new seal (18094-02A) is orange in color while the 2002-06 one is black rubber (18094-02).

The heads still require porting and polishing to raise the horsepower, which does not increase substantially from the Twin Cam 88 to the Twin Cam 96 even with eight more cubic inches and higher compression of 9.2:1. Nevertheless, the head is a big improvement over what went before.

Installing 2007 TC96 heads on earlier 1999-2006 engines will result in much improved performance. However, it is not as much performance as could be gained by massaging them further.

A major Twin Cam engine deficiency has always been

insufficient head airflow. The head design is the best since 1903. Quality is excellent. It is just that the ports and valves do not deliver enough air to allow the engine to be what it could be. H-D addresses airflow to some extent in the TC96. In my opinion, it is not enough.

I hope that improving technologies can deal with emissions dictates.

Porting and polishing is much more than removing port obstructions and port re-sizing, if necessary. Cutting the valve seat is the most important aspect of increasing flow in a head. Valve seat fitment relative to the combustion chamber on one side and its relationship to the port on the inside is also crucial to maximum flow.

## Cutting the Valve Seat

Valve seats if cut at one angle only, give demonstrably poor airflow results when compared with a standard three-angle valve cut that produces a satisfactory production motorcycle outcome. Five-angle seat cuts produce far better airflow increases. If a seven-angle cut is attainable, this will give the best results.

## Interference Valve-to-Seat Fit

Normally the base seat cut in the head where the valve head actually touches and seals the combustion chamber is at while the valve face cut is at an interference fit of about

## Valve Seat Width

The adjacent cuts to the heads valve seat will be at and This produces the three-angle cut. With these two contiguous

cuts, we can determine the width of the actual valve to seat area. Cutting the neighboring and valve seat cuts deeper will narrow the -center contacts. Cutting the adjacent and valve seat cuts lighter will allow a wider -center contact. Cutting the contact cut deeper will widen it.

Moving the Valve Seat to Valve Contact Area

The seat contact area can move out or in by deepening one adjacent cut relative to the narrowing of the other to place it where desirable on the valve face. Placing the valve contact position to the optimal seat contact location will increase flow. There is no set placement as port design and combustion chamber shape along with valve-head size all combine to determine the location of the optimal valve-to-seat contact on the rounding radius of the valve seat

## Positioning Valve Seat for Best Flow

Generally, for best flow, place the seating area close to the rim of the valve. However, do not put the seat right at the rim, as this will lead to a premature compromise of the contact area from the searing combustion heat. This contact cut is very important as a wide contact area may have more longevity before re-cutting is again required but will not flow as much air as a narrower cut. A very thin contact area will produce the most flow but may not have satisfactory longevity before a rebuild is necessary. Very narrow cuts have use on a race bike. Therefore, on a street bike, we need a compromise valve-to-valve seat contact area, both to ensure good airflow and engine longevity although the choice is up to the rider after consulting with the mechanic.

## The 5-Angle Seat Cut

A five-angle valve seat cut adds additional and cuts. If we look at the seat from one side to the other, the cuts will be in the following sequence: and If we look at it from the opposite side, the figures reverse.

The 7-Angle Seat Cut

The seven-angle cut will add two cuts, one at either end, say and further rounding the seat. The thinking mechanic will decide on the angles for these two cuts depending on the contour of the head port approaching the seat and the contour of the combustion chamber moving away from the other side of the valve seat.

The key word is A rounded valve seat promotes the easy flow of air removing impediments to the volume and velocity of flow.

## Valve Seat Fitment

The valve seat is a separate part pressed into a cutaway in the head. This is necessary because the seat is made from a very hard steel alloy in order to withstand the pounding of the valve returning to the seat thousands of times per minute, not to mention the extreme heat that seeks to burn away the

seating area. The thin seating area is subject to the searing heat because of its thinness and therefore vulnerability.

The head composition is from a much softer aluminum alloy. The alloy could not withstand the pounding or the concentrated heat.

The seat abutment where it contacts the head in the port, whether it is the intake or the exhaust port, is also crucial to uninterrupted airflow. If there is an overhang, it must be radiused. If an under-hang, the seat abutment requires contouring.

CVO 110

The heads, front OEM 17250-07, and rear OEM 17248-07 have big expanding intake ports similar to a Shovelhead to accommodate bigger intake valves. The valve seats are thinner

in depth and width as the following chart will demonstrate.

Comparing TC88 and CVO 110, Valve Seat Dimensions

The Screaming Eagle heads on the CVO 110 only have a 3-angle valve seat cut. Optimally we want a 7-angle cut but many times, we cannot do this because the seats have already been radiused out at the Factory. However, a 5-angle cut to each valve seat face will further round the entranceway for the incoming fuel charge and initial departure port for the exiting exhaust gases.

The Flow Bench

The flow bench will teach the student the physics of

airflow as the valve opens and closes. Hundreds, if not thousands, of hours of work on a tool called a flow bench will teach the prospective porter what works and what does not. Measurement of airflow is in cubic feet per minute (cfm). The volume of air passing through the port changes as valve lift or valve opening increases or decreases. This gets complicated very fast as the reader can now discern a relationship between cam lift and airflow.

We need to ask questions. Where do we want maximum airflow that leads to maximum power? We now begin to design the power curve, where power comes on and at what rpm it will begin to taper off. Merrily cutting and grinding away just does not make it!

Many reworked heads that I have seen actually reduce airflow rather than improving it. This is no area for amateurs,

as porting heads with the valuable assist of the flow bench requires skill, patience and some art.

The concept of airflow is a lot like that of electricity for mechanics and riders alike. Both are invisible. What cannot be seen becomes complex. Therefore, many cannot understand and grasp the simple complexities of both. Electricity powers the engine management system, charging, and lighting. Air powers the bike by mixing with fuel allowing it to burn, creating heat, and thus energy to drive the bike forward. We need to measure airflow. We need to know when we make performance and/or efficiency changes if airflow increases or decreases.

## Visualizing Air Flow

I go through this analogy of water and air many times. I realize I am being repetitive but I did not understand these

concepts myself until examining water flow characteristics.

Thus airflow is easier to understand if the reader visualizes a river and how water acts as it makes its way downstream. The following examples are simplistic but demonstrative.

A narrowing will increase flow, as the same volume of water must pass in the same amount of time. The weight or pressure of the water behind will increase the speed of the water to accomplish this. Similarly, a widening will cause the water to slow down as the pressure from behind eases. I have just described Bernoulli's Principle, which has to do with the movement of gases or liquids.

If observing a rock in the middle of the stream, watch how the water moves around it. Look for shapes that allow the water to flow easily over and around it. Then study shapes

that impede flow. It does not take long before learning how to shape the rock to improve flow.

It is very interesting and predictable, as is head airflow.

Let us get back to the river to demonstrate that a rough intake port tract does not have that much effect on mixture of the fuel and air. The water flowing on the banks or bottom may have some minor turbulence as it interacts with the rock or sand. This turbulence, if there is any, does not transfer very far, if at all, into the main stream of water movement.

The same is true of the intake walls affecting air/gas mixture. There is a little mixing where the interaction occurs, but it does not affect the main body of laminar air. Interestingly, this also applies to oil flowing through the oil cooler. Only the outside layer in contact with the channel

cools. The inside oil in the center flow does not. In this case, I want a turbulator effect to occur. The rider will wisely choose an oil cooler like those designed by Jagg (www.jagg.com). The inside passage of these oil coolers have obstructions built in to turbulate and mix the oil so that all of the oil has an opportunity to dissipate its heat. Most oil coolers lack this simple and effective mechanism.

Let us get back to head flow. Squish bands in the combustion chamber produce the turbulence we need for a high percentage of burn. The more combustion chamber turbulence, the better the mixture will be in correct proportions. This leads to a higher percentage of burn leading to engine efficiencies, fewer emissions with more horsepower as a very welcome by-product.

Coordinating Head Flow and Lift

When head designing engineers and flow bench technicians measure cylinder head port cubic-feet-per-minute (cfm) airflow, they do it at differing valve lifts. The exercise becomes how much air will flow when the valve is open at say, .510” lift. Head port design as well as with porting and polishing existing heads flow cfm’s at different rates for differing valve lift openings.

Therefore, port design determines optimum lift for the cam.

As a simplification, installing .650” lift cams in an engine, whose heads flow maximum cubic-feet-per-minute airflow at .550” valve lift may be counterproductive depending on cfm drop-off rate after maximum flow for the remaining .100” of valve lift. Stock TC88 or TC96 heads can hardly handle cams with either the lifts compared above when they match cams with lifts of anywhere from .471” to .484”. This is another reason why headwork is crucial in performancing the Twin

Cam.

## Comparing Flow Bench Results

There are many variables from one flow bench to another. Operator errors such as not recalibrating the flow bench for new testing can result in errors of five or more cfm readings. Leakage from fixtures will also give wrong cfm flow results. The air inlet on the flow bench should be radiused with all head testing. If the inlet is sharp, airflow disrupts and can easily vary with up to 25% less flow when compared with say, the same head using a radiused inlet. Correct radiusing will ensure maximum laminar airflow.

Laminar defines as a smooth layered flow of a gas, in this case air, in which neighboring layers do not mix, and flow at different velocities.

Barometric pressure is a major factor as air density varies with both altitude and the weather conditions. Air is denser when cool. Therefore, there is more air in the same amount of space in colder weather than hot. This is why an equalization process of barometric pressure must occur when comparing ratings for head flow or else effective and honest comparisons will be void.

In my opinion, pinpoint accuracy in comparisons is not possible but it is easy to compare fairly but not very accurate flow rates from one bench to the other if all testing parameters are the same. We can compare if using among other things the same attachment to affix the head or filter, as the case may be, to the same model flow bench and use the same pressure in the test.

This is where the consumer gets into trouble and may not make the right decision in choosing a particular product. Competing products like performance heads usually test on different flow benches. They may not have the same affixing attachment and may be test with different pressures producing wildly different and conflicting results that are not comparable at all. Using other supposedly comparative tests but with different parameters skew the results out of control making one product falsely superior or inferior to another.

However, if testing the products within the same specific parameters results will be roughly accurate and thus produce valid comparisons.

## Volumetric Efficiency

Volumetric efficiency or the biggest percentage of cylinder fill will occur just as the valve is closing. In almost all cases,

published flow bench tests will advertise the best result. Unfortunately, the next advertised test might show a result using a different valve lift opening. Who is to know, which head really flows the most air? It is imperative to know that all testing parameters are the same or else the figures will be useless.

Test Pressure

Flow benches like SuperFlow (www.superflow.com), compare two air pressures; one is a calibrated pressure across a known orifice or opening; the second is a constant test pressure fed through a breather, carburetor or in this case a head. The air feeds via an air pump much like that of a household vacuum cleaner. This continuous and constant test pressure drops across the opening. Calculating the ratio of the pressure drop relative to the calibrated known pressure

determines volumetric flow, which is fancy lingo for the amount of air measured in cubic feet per minute, (cfm).

Different testing facilities and manufacturers use different test pressures making comparisons difficult for the consumer. A column of water called a manometer indicates and validates the test pressure chosen. The height of the water in the manometer column visualizes the invisible test pressure. The suction of the flow bench drawing air through the test item such as an air filter also pulls on the water in the manometer's column. The height of the water determines and reflects the air test pressure.

One atmosphere, which is the gravity weight of air at sea level, is equal to a column of manometer water of 403 inches. Of course, using a manometer of this height is awkward. Generally, use of between about one-and-a-half inches and twenty-eight inches of water are typical test pressure choices.

Measurement of actual airflow is via a second manometer measuring the pressure drop. If there was a standardized manometer test pressure that different manufacturers and technicians could agree on, the test results, everything else being equal, become comparable.

It is imperative to maintain the same test pressure as indicated in the manometer inches of water during the entire test procedure or results will again skew and become useless. This stuff sounds complicated but its foundation bases on high school physics.

## Calculating Flow Bench Test Pressure Conversions

cfm = cubic feet per minute

Formula: Flow = (new flow pressure ÷ known flow pressure)2 x cfm

Example: 150 cfm @ 10”, convert to 28”

Flow = (28 ÷ 10)2 x 150

Flow = (2.80)2 x 150

Flow = 1.675 x 150

Flow = 251.25 cfm at 28”

Lighter Valve Train

Valves and guides are thin stem since 2005. Traditionally

Harley-Davidson Big Twin valves have a three-eighths inch valve stem. This now reduces to five-sixteenths-of-an-inch. This was an important advance since the valves are obviously lighter.

Valve trains in pushrod-operated engines like all traditional Harley's, the V-Rod being the first exception, are heavy affairs to move up and down with speed as in rpm's. Their weight limits their rpm capabilities.

Overhead cams like those used in the V-Rod that actuate the valves directly are capable of high rpm's.

Any time we can lighten the valve train, the engine will be capable of slightly higher rpm's and therefore more power. There will also be the added advantage of increased parts longevity. A lighter valve train means that the valve-spring

seat-pressure does not need to be as great to return the valves to their seats as the commanding valve lobe rotates to a lower position. This relieves pressure on the valve train allowing parts to last longer.

Heads and component parts coordinate for a lighter valve train plus higher flow. Pistons are shorter to accommodate the three-eighths-of-an inch increased stroke in the TC96.

## Edelbrock Performer RPM Heads

Edelbrock (www.edelbrock.com), a world-renowned automotive performance company lends its expertise to the Harley-Davidson performance market.

They use swirl-polished steel valves along with radius valve seat cutting for superior flow characteristics. CNC ported Performer heads accommodate up to .650” valve lift

with installed 175 lbs-seat-pressure performance springs.

Edelbrock claims and I believe them; their heads have high-velocity airflow that results in quick throttle response. All Edelbrock heads necessitate their matching rectangular port manifold to mate with their head's intake port rectangular shape. This shape promotes laminar flow and thus higher velocity airflow. The floor or bottom of the rectangle slows airflow as it goes around port corners to keep in pace with the faster moving upper flow that must traverse more distance. This design keeps airflow equal and thus smoother reducing the turbulence that slows down and air/fuel charge.

The dual quench combustion chamber promotes more turbulence during the compression stroke to assist atomization for a higher percentage burn.

The D-shaped exhaust port is an important ant-reversion feature.

## Edelbrock Head Twin Cam Fitment

## S&S Super Stock Performance Heads

“On the Superflow 600 flowbench our 89cc S&S Super Stock heads flowed 264 cfm @ 25” water compared to 182 cfm @ 25” water for stock heads from a Harley-Davidson Twin Cam 88 engine…S&S heads allow the use of cams with as much as .640” of total lift.” 1

The 88, 95 or 96-inch engines can operate without compression releases. However, the machining allows the use

of S&S compression releases if needed. If so, the compression releases require the use of S&S head wafers.

The 79cc combustion chamber yields 9.1 to 10.3:1-compression ratios. The 1.940” intake valves assist higher velocity of fuel and air.

The 89cc heads do not perform until about 103 cubic inches and will accommodate up to a 4.000” cylinder bore. Compression ratios with the 89cc heads for 100 cubic inch to 116 cubic inch engines vary between 9.2:1 and 10.4:1.

Both 79cc and 89cc chamber heads can increase compression further by planing the gasket surface up to an additional .060”.

Of course, it is crucial to check valve–to-piston interference.

# Screamin’ Eagle Heads

The H-D aftermarket heads are of the same superb construction as the stock TC88 and TC96 heads.

Screamin’ Eagle has made many offerings of differing heads since 1999 for Twin Cam performance. In my opinion, they are all-good performance heads when combining with the appropriate pistons and coordinated with other complementing performance parts.

I believe some are no longer in production because of emissions laws. The old disclaimer that these products are for off-road use only or race only is receiving challenges from the EPA.

I will describe some of them, but for me to say, which is the best is not fair because they will all work well as part of a coordinated team with specific goals for the type of performance desired. All of the head combinations listed below require at least an ECM calibration if fuel injected when using prescribed Screamin' Eagle components only. Fuel modifiers are a better alternative, in my opinion especially if not following exact components for the ECM calibration.

Carburetion will need modification also. I cover these subjects at great length in Chapter IX: Fuel Delivery.

Screamin' Eagle Pro Twin Cam 110+ Heads

Harley-Davidson is entering the realm of big inch engines with this offering of heads (black highlighted 17071-08, silver highlighted 17072-08), for 110c.i and bigger cubic inch engines for 2007 and later models. Most people I know refer

to the silver heads as blond. They can retrofit back to 2006. Combustion chamber size is 95cc with fitment of a 4.000” bore comparing to a stock TC96 bore of 3.750”, a .250” diameter differential.

These heads have fitment for automatic compression release (ACR) to offset starting problems with the 10.5:1 compression ratio.

The very big 2.120” intake valve and a larger 1.623” exhaust valve can open up to .650” valve lift.

Screamin’ Eagle Pro Twin Cam ACR Performance Heads

The part number for black is 17165-08 and for silver is 17166-08 and fitment is for 110+ cubic inch engines for 2007 and later models. They can retrofit back to 2006.

Combustion chamber size is 95cc with fitment of a 4.060” bore comparing to a stock TC96 or TC88 bore of 3.750”, a .310” diameter differential. A 4.000” bore with 4.375” (4 ⅜”) stroke equals 110 cubic inches while increasing the bore to 4.060” with the same stroke will equal 113 cubic inches. This stroke of 4.375” is the same as that for the TC96 while the TC88 has a 4.000” stroke.

These heads have fitment for automatic compression release to offset starting problems with the 10.5:1 compression ratio.

The 2.080” intake valve and a larger 1.630” exhaust valve can open up to .650” valve lift.

Screamin’ Eagle Pro Twin Cam MCR Performance Heads

The part number for black is 16952-08 and for silver is 16953-08 and fitment is for 3 ⅞” (3.875”) bore.

MCR is the acronym for Mechanical Compression Release (32076-04). Mechanical releases manually operate by the rider while automatic compression releases operate automatically. Combustion chamber size is 76cc with a 10.5:1 compression ratio. These heads will accept up to .575” valve lift cams and can rev to 6500 rpm.

A personal comment from me would say that taking a Harley-Davidson pushrod-operated engine to 6500 rpm is flirting with danger. Some Screamin’ Eagle heads have rpm potential of 7000 rpm but I am never taking my bike to these levels. The powerband will have peaked before these rpm levels anyway.

Fitment is for 1999-2005 Twin Cams that require an additional purchase of two intake flanges (26993-06) and for 2006 to present models that can use their stock intake seal flanges.

Screamin' Eagle Pro Twin Cam 103+ Heads

The part number for black is 17071-03B and for silver is 17072-03B.

The intake flanges for Twin Cam heads changed circa 2006. These heads allow the new flanges. Therefore, they will accept the 50mm electronic fuel injection induction module, which is bigger than any stock one including the CVO 110 by at least 4mm depending on year.

The combustion chamber is 98cc with a big 2.080" intake valve and a 1.630" exhaust valve. They will allow valve lift of

up to .650”. Head design is for 103+ cubic inches on at least 3.875” bore barrels or larger. This bore is 3 ⅞”, the big-bore sizing for both the 95 cubic inch version of the TC88 and the 103 cubic inch version of the TC96.

These heads are too big for the 95-inch engine. Flywheels need to upgrade to 4 ⅜” to achieve 103 cubic inches with the 3.875” big-bore. Another way is to increase the bore yet again to 4.000” or 4.060” to achieve the cubic inches that these heads need.

These heads approximate a hemi-head design as opposed to a bathtub one. However, there is still a squish band, which may elude an unpractised eye. Piston valve-pocket-cut-outs interrupt the hemi-squish band that looks like the beginning of a piston dome that reverses itself in the piston dome center. This squish mates with an angle at the beginning of the

combustion chamber to mate with a similar angularity on the piston dome.

I suspect this design combats valve shrouding whereby the valve gets in the way of incoming and exiting gases.

The reverse dome lowers the compression ratio to 10.5:1. I am not a big fan of reverse dome pistons as I do not feel that they encourage the best combustion flame travel possible.

The heads have a 7000-rpm capability. The head machining accepts compression releases. The H-D part number is 32076-04. Fitment is for 1999 and later Twin Cams with 3 ⅞" big bore TC95 or TC103 kits; 1999-2005 Twin Cams require intake flanges (26993-06).

Screamin' Eagle Pro Twin Cam Ported HTCC Heads

These heads are the currently the highest flowing head from the Motor Company. Therefore, this is one of my favourite head choices. I have them on my 2007 Streetglide with the 50mm EFI induction module with the late-style intake mounting flanges.

HTCC is the acronym for High Torque Combustion Chamber. The piston and head design use the hemi-squish band. I would rather have the bathtub head design (black 16933-99B, silver 16926-99B) with the 95-cubic inch style HTCC pistons (22439-00A), but my Mama says that I cannot have everything in life.

The 10.5:1 compression heads with an 84cc combustion chamber with a 2.000” intake valve and a mating 1.623” exhaust valve, accept up to .650” valve lift cams and come in black (16925-02B) or silver, (16934-02B). Their design is for

103+ cubic inches on a minimum 3.875” bore. The heads have a 7000-rpm capability.

The head machining accepts compression releases (32076-04). Use the Screamin’ Eagle manifold (29635-99), in conjunction with these heads.

Fitment is for 1999 and later Twin Cams with 3 ⅞” big bore TC95 or TC103 kits; 1999-2005 Twin cams require intake flanges (26993-06).

Screamin’ Eagle Pro Twin Cam HTCC Heads

The 10.5:1 compression heads with an 84cc combustion chamber with a 2.000” intake valve that accentuates low-rpm torque and a mating 1.550” exhaust valve, accept up to .575” valve lift cams and come in black (16933-99B) or silver, (16926-99B). Their design is for 95 cubic inches on a

minimum 3.875” bore. These are my favourite heads matching with HTCC pistons (22439-00A). However, they do not fit every application such as my current 2007 103” FLHX.

The heads have a 6500-rpm capability. The head machining accepts compression releases (32076-04). Use the Screamin’ Eagle HTCC manifold (29634-00), in conjunction with these heads for correct head port matching.

The 95-inch HTCC heads have a hemi-multi-layer squish band arrangement. The intake port is a D-shape as opposed to round, thereby promoting a high velocity fuel charge. The intake port has a step in the bowl area to minimize intake reversion while the intake valve is still open when the piston begins ascension on the compression stroke. The exhaust port has an oval shape to assist in exhaust gas scavenging with a smaller exhaust valve that takes advantage of Bernoulli’s

Principle for velocity of exiting gases.

Fitment is for 1999-2006 carbureted models plus 2001-2005 EFI Softail, 2002-2005 FLT models and 2004 to 2005 EFI Dyna models with 1550 cc big bore kits.

Screamin' Eagle Pro Twin Cam Performance Heads

These heads utilize Evolution style port shapes with Twin Cam 72cc bathtub combustion chambers. The heads accept up to .575" valve lift cams Therefore, they use a TC88 or TC96 flattop piston.

Do not use the higher compression cast pistons that normally increase TC88 compression to 10:1. A compression ratio of 9.5:1 on a TC88, 10.0:1 on a TC95 or high compression forged pistons will bring both configurations to 10.5:1.

Fitment (black 16952-99C, silver 16953-99C) is for 1999-2006 carbureted models plus 2001-2005 EFI models; 1999-2005 Twin cams require intake flanges (26993-06).

Fitment (black 16952-06, silver 16953-06) is for 2006 and later EFI models.

The head machining accepts compression releases (32076-04).

Performance Head Comparisons

3.875”

*Compression ratio is 9.5:1 on TC88, 10.0:1 on TC95, although high compression pistons will bring both configurations to 10.5:1.

** Compression ratio for 79cc heads is 9:1 on a TC88, 9.6:1 to 10.3:1 depending on piston on the TC95, 10.1:1 on 96c.i and 100 c.i; 10.4:1 on 106c.i; 10.7:1 on 107c.i and 11.5:1 on 116c.i. Compression ratio for 89cc heads is 9:1 on TC88, 8.3:1 to 8.7:1 on TC95, and 9.2:1 on 96c.i. Therefore, we see that the 89cc heads are not appropriate for these smaller engines. Compression ratio for 89cc heads on a 100c.i engine is 9.2:1, 9.8 on 106c.i, 9.7:1 on 107c.i and 10.4:1 on 116c.i.

*** Compression ratio depends on piston use with a 10.25:1 ratio utilized as a power package.

## The Exhaust System

Many manufactured exhaust pipes today are too skinny, too fat, too long or too whatever to produce good gains in horsepower. There is even the odd exhaust system touted for performance gains that will cause a decrease in torque and horsepower over the restrictive stock ones.

There is no one-production exhaust that I or anyone else can say is the best Many exhaust pipes are the best in certain specific areas. Some pipes are versatile because they are tunable.

However, I can say that there is a best production exhaust

system for a specific engine and weight-to-power ratio, which analyzes total weight of the bike and rider to engine performance and a particular riding style. In general, I can probably narrow the search quite effectively to a two-into-one collector system with a bigger muffler and restrictor pipe.

Those designed purely for looks or sound is suspect in the performance world. In most instances, noise does not equate with power. Actually, the only situation that comes to mind where extreme noise and mega-power co-exist is with top fuel Harley dragsters. On street bikes there will be some noise with excellent power gains but not always, as much as one might suspect.

Some exhaust systems are light years ahead of others because their design adheres to sound principles of physics. It has been common knowledge for almost one hundred years that muffler volume should be at least ten times that of the cylinder that it serves with a restrictor pipe the same size as

the header pipe exiting the muffler. This will quieten the exhaust noise with no loss of power.

Of course, today we know that we can increase power with correct header pipe diameter size, length, and muffling. Now we know that the muffler has to be large in order to work optimally.

Skinny mufflers had better have some happening new technology going on if they want to compete with larger expansion chambers. Style traditionally interferes with performance. However, some manufacturers today will achieve a stylish fat appearance by building a cosmetic shell around a narrower performance header pipe.

The header pipes join to the engine heads and continue down to the muffler(s). For some inexplicable reason, fatter is

more beautiful on a Harley-Davidson, perhaps another rebellious trait that draws so many to the freedom machine.

This type of design, where cosmetics shroud hard working efficiency combines form and function and is truly wonderful. Technology and design improvement abounds; I give an example from the venerable Vance & Hines.

## Performance Exhausts

### The Power Chamber

The Power Chamber is an innovation for Harley-Davidson aftermarket exhausts brought to us by Vance & Hines (www.vanceandhines.com), in 2001. The Power Chamber connects both staggered dual mufflers internally to increase volume to improve the scavenging of each cylinder for more efficient combustion. This develops more torque and thus

horsepower while keeping a formerly less efficient exhaust design. This cool originality delivers good looks and power at the same time.

## The Harley Styling Enigma

It is strange that we like a fat header pipe that in most cases will reduce power and yet we hate a fat muffler that usually increases power.

Modern exhaust designers are dealing with this ugly issue by making the collectors sleek and stylish as opposed to the tin can look that you see on Japanese racing bikes. Check out the exhausts used on some model Buell's; they are all function and power producing. They may also hide under the bike for aesthetic reasons. Buell exhausts are state of the art. They meet all performance, noise, and emissions requirements.

However, while they may be good-looking to a sport bike rider, they sure fail with the average H-D rider.

## Screamin' Eagle

Screamin' Eagle (www.harley-davidson.com), exhaust slip-on mufflers maintain the stock headers and emissions fittings like the crossover pipe and oxygen sensors and bungs as do some other aftermarket slip-ons.

In my opinion, the older Screamin' Eagle mufflers discontinued a few years ago because the off-road disclaimer began to wear a little thin. In fairness, I think the EPA and CARB will go after the major manufacturers first before enforcement looks at dealers, aftermarket shops and finally the consumer. I also think H-D realizing future constraints, voluntarily developed a new performance exhaust line in

advance of enforcement that complies with existing laws.

This is both intelligent and progressive at the same time. The new Screamin' Eagle exhausts meet mandated decibel (dB-A), noise ratings and current emissions controls. The new slip-on mufflers are for specific models. They have never been interchangeable primarily because of muffler bracketing.

Pleasant muffler sound noise accomplishes by changing the diameter of the muffler itself and with differing types of baffles filtering out higher frequencies. The TC96 Softail, Dyna, and Sportster mufflers are 3.5" in diameter while FLT mufflers are either 3.5" or 4.0" diameter. It is no mistake that stock FLT Baggers make more power than other stock model Twin Cams; the simple basic reason is that their mufflers have more internal volume.

The EPA is not destroying muffler performance functionality so much as causing R&D (research and development) investments to control mutually harmful emissions, noise pollution, but also the development of superior performance exhausts.

Harley-Davidson mufflers will give a few extra ft-lbs of torque and perhaps four to five extra horsepower without ECM recalibration. Adding their air filter, which helps complete the breathing equation will almost double the muffler performance increases.

The Stage I recalibration may or may not be necessary since many TC96 Twin Cams already come with a slightly rich AFR from the Factory. If not doing the recalibration, make sure to check the AFR is not too lean with intermediate and full throttle spark plug color readings. I describe this process in the next chapter, Chapter IX: Fuel Delivery, section, Spark Plug Color Tuning.

Python

Python (www.dragspecialties.com), staggered duals have always been excellent for performance when comparing to the older availability of two separate pipes also known as staggered duals.

Python's were the first production exhaust that I am aware of that incorporated torque cones also known as torque or tork valves and more properly known as anti-reversion valves or cones in the top of the header pipes. Anti-reversion is descriptive in that these reverse shape cones help deflect returning kinetic wave pulses back out the exhaust instead of re-entering the combustion chamber to pollute the virgin charge of fuel. Torque valve is also very descriptive because these cones increase engine torque.

Moreover, Python staggered duals came in differing performance versions for stock or modified engines and for bigger displacement engines. The newer Python staggered duals have standard 1 ¾" headers feeding into 2 ⅛" muffler bodies with a hidden crossover pipe that increases performance throughout the rpm range. They are now available with 02 sensor bungs also. The baffles are removable. Quieter baffles are available that coincidentally will increase low-end torque.

Two-into-one exhaust systems will produce the most performance gains. The Python 2-into-1 exhaust has a big 3 ½" muffler and is available with 02 sensors. Riders may not like the look of big mufflers but should remember that the muffler should have ten times the volume of the cylinder(s) it serves.

Once knowing this little-known fact, the reader can immediately begin eliminating many exhaust systems from their performance lists.

## Cylinder Head Anti-reversion Exhaust Ports

One of the many great design changes in the Twin Cam 88 engine is the cast in anti-reversion step in the head where the exhaust ports meets the exhaust pipe. In fairness, the credit should go to Edelbrock (www.edelbrock.com), followed by Patrick Racing (www.patrick-racing.com), heads, which were the first to my knowledge to use this anti-reversion design. They first did so, on their exhaust port exits of their Evolution fitment style aftermarket heads. S&S Cycles, Inc. (www.sscycle.com), and Ultima (www.jirehcycles.com), also incorporates anti-reversion steps on all their heads many of them designed well before the advent of the Twin Cam.

The exhaust port exit normally is round in stock Shovel and Evo heads and now with the TC96 heads just like the exhaust header. The modified shape of the exit port now found in stock TC88 Twin Cam's and the aftermarket Evo and Twin Cam heads assists the exhaust valve in reflecting back returning exhaust pulses.

Thunderheader Exhausts

Thunderheader (www.thunderheader.net) is a division of Rich Products. One only has to see the big collector or mufflers to realize the reason that some of these produce the best torque /horsepower results given certain engines and combination of support hi-performance parts. Thunderheader with its unique sound has become a street-performance-enthusiast favorite. Its distinct look and length oozes performance.

Hooker Exhausts

Let us look at some exhausts like Hooker, (www.holley.com), who makes a wide range of stylish exhaust s for Harley-Davidson. This premiere company, part of the Holley dynasty divides each performance styling into two sizes or classifications; exhaust systems for 80 to 95 cubic inch engines and ones for 96c.i and larger displacements.

Add to this that some Hooker exhaust systems like their Bagger mufflers and 2 into 1 collector have a means to control the volume of backpressure. They are tunable. Tunable exhausts fit a wider range of engines and produce optimal power within a useable power band. Hooker's have a tunable, rotating disc that can turn for different power producing engines.

As the reader will see, SuperTrapp takes this tuning component a step further.

SuperTrapp

SuperTrapp (www.supertrapp.com) is a well-known performance exhaust manufacturer that has been around a long time who also makes the performance Kerker line.

SuperTrapp has a tunable exhaust. Now, this is big for the performance oriented, as SuperTrapp exhausts can actually move the power band up or down the rpm range. A progressive fellow, Paul Moller invented the Tunable Disc Technology way back in 1971. Riders can easily add or subtract theses discs from the end of the muffler to customize their performance needs and/or simply achieve a pleasing sound to their ear.

Adding more discs will increase the exhaust outlet and decrease backpressure. The addition of more discs will widen the Powerband at the top end for highway riding. This will also increase the exhaust tone. The tuner must be aware that the air-fuel ratio will lean out and fuel adjustments may be necessary. Removing discs to decrease the exhaust opening will increase exhaust backpressure. Disc removal will narrow the power band. This creates more low-end torque. Furthermore, it will decrease the exhaust tone and the air-fuel ratio (AFR); will lower (rich condition). An overly rich AFR will decrease power so it may be necessary to lean out the air-fuel-ratio.

Information is Power

There are many more exceptional exhaust systems in the marketplace that space does not permit the exploration of

here. If a particular exhaust goes unmentioned in this book, it does not mean any negativity should attach.

However, there are many more lousy exhausts in terms of performance than good ones. After absorbing the information in this and other chapters, the reader will be more than capable of making their own competent analysis and therefore, choice.

## True Duals

In my humble opinion, True Duals, the best known is Rinehart from Rinehart Racing (www.rinehartracing.com), which are a current fad on FLT Baggers, cannot make good power gains by eliminating the crossover equalization pipe standard of Baggers for a half century unless other technology incorporates in the design.

Rinehart Racing whose name derives from Gerald Rinehart of NASCAR fame and Bub Enterprises (www.bub.com), currently challenging land speed records at the Bonneville Salt Flats makes the most popular True Duals. They are no slouches in the performance game. Their True Duals are strikingly beautiful.

The much-maligned stock crossover pipe on the Harley Baggers serves a good performance function as it attempts to equalize pipe balance. The crossover balance tube assists in bottom-end and mid-range performance while its elimination will move the Powerband up into mid-range to top-end performance

I recommend leaving the stock header pipes unless increasing displacement substantially and use a tunable big volume muffler like Hooker or SuperTrapp for performance.

Both Rinehart Racing and Bub manufacture quality products and many of their exhaust systems combine beauty and performance.

## Power Equals Breathing

Modern Harley-Davidson's, certainly through later Evolution years and now the Twin Cam have breathing restrictions.

### Air Equals Power

Gasoline needs air in certain ratios to produce the highest percentage of burn, producing heat that expands.

The expanding heat creates the pressure to drive the piston down on its power stroke to produce the twisting force, torque that turns powertrain components and ultimately the rear wheel.

Horsepower is only an arbitrary figure that calculates by multiplying torque times rpm.

The stock cam(s) and air cleaner will limit the performance of your engine, even with the installation of a high performance exhaust system.

The engine is like a human lung. The lung requires the ability to breath in as much air as it needs through the air breather, induction system, and intake port with intake valve. The intake valve opens longer with cam duration and further

with cam lift allowing the ingestion of more air. The exhaust tract and exhaust valve in combination with a performance exhaust allows the spent gases out of the lung so the cycles can repeat and bring in fresh oxygen.

However, the exhaust also creates returning pulses that it must discourage from re-entry into the lung. Furthermore, it creates the pressure of momentum to assist in vacuuming additional clean air into the lung. The exhaust system will have a very dramatic effect on how an engine operates throughout its rpm range. The exhaust system will affect torque, peak horsepower, and rpm as well as influencing throttle response, gas mileage, vibration, and engine temperature.

Engine temperature is a big deal now that Harley-Davidson uses closed loop electronic fuel injection also known as a

feedback system. Oxygen sensors send exhaust composition analysis via electrical pulses to the electronic control module, which regulates fuel delivery and ignition timing to create Stoichiometric 14.7:1 air-fuel-ratios during hot idle and cruise situations. The sensors will feedback information constantly to achieve this lean and hot air to fuel proportion. The exhaust system can assist in lowering injurious engine temperatures or it can exacerbate the problem, further adding unwanted heat.

It is clear that the addition of a high performance exhaust system alone will not achieve maximum torque and thus horsepower. The engine must have the proper components necessary to produce maximum horsepower output. Simply it needs to breathe.

With the installation of the breathing components, which are a high performance cam(s), carburetor/EFI, air cleaner,

ignition and exhaust the expectation of huge percentage increases of torque and horsepower over stock are a given. How much, well who can say without more specific information but a 30% increase is easy to achieve over stock. Headwork, which will increase breathing substantially, in combination with supporting performance parts that assist in the breathing volume, can surely take the increase to 50%.

Increasing cubic inch displacement and/or utilizing forced induction will have different exhaust system demands for performance but can take percentage increases of 100% or more over stock.

## The Catalytic Converter

Exhaust systems perform several functions such as removal of spent gases, quietening noise and reducing emissions. They also allow cooling down of the gases that would otherwise

burn out the exhaust valves without header pipes.

## Exhaust Composition

Exhaust gases comprise of unburned hydrocarbons (HC), carbon monoxide (CO), carbon dioxide (CO2), nitrogen oxides (NOx), sulfur dioxide (SO2), phosphorus (P), and sometimes very small parts of the heavy metals molybdenum (Mo) and lead (Pb).

Catalytic Converters have been in use on California Harley's for some time now and may spread to the other 49 states before or after 2010 when extra restrictive EPA emissions standards become law.

Catalytic converters use metals like platinum, palladium and sometimes nickel to react with toxic exhaust gases and convert them to if not harmless, at least less dangerous

substances.

Global Emissions Systems (www.gesi.us), has reinvented the catalytic converter with a newer technology basing on current methodology to reduces up to 100% of Hydrocarbons (HC), up to 99%Carbon Monoxide (CO), and reducing Nitrogen Oxides (NOx) and carbon dioxide in gasoline combustion as well as other fuels. GES uses slightly different metallurgy in different quantities to achieve their emissions goals. The cost is around $1000.00 to $2000.00 U.S. per vehicle installed. The GES converters measure about six-inches by eight-inches for a car, so are very adaptable.

## Exhaust Pulses

To gain a better understanding of how mufflers and headers do their job, the reader must familiarize with the

dynamics of the exhaust pulse. Exhaust gas does not come out of the engine in one continuous stream. As the exhaust valves open and close, the exhaust gases will flow with a burst of pressure, and then stop, and then allow another pressure burst again as the exhaust valve opens. Of course, this process occurs thousands of times per minute.

For an exhaust pulse to travel, the leading edge must be of a higher pressure than the surrounding atmosphere. The body of a pulse is very close to ambient pressure while the tail end of the pulse is lower than ambient pressure producing negative pressure or a trailing vacuum. This pressure differential is what keeps a pulse moving. Therefore, exhaust gas is actually a series of pulses. An H-D exhaust can make up to fifty pulses per second.

The low-pressure tail end of an exhaust pulse will most

definitely attract the high-pressure leading edge of the following pulse, effectively scavenging it along. The header tuning in a performance exhaust specifically allows the exhaust pulses to line up and scavenge each other forward.

Exhaust scavenging also sucks in the fresh intake air and gas by the fast exhaust exit of the combustion chamber. The trailing negative pressure of the exiting spent gases assist greatly in drawing in the new charge at higher rpm levels.

However, as engine rpm changes up-and-down, the lining up of exhaust pulses also change. Therefore, the header pipe design will be for certain rpm ranges. Stock mufflers are so restrictive that a side benefit to them is that they ram the exhaust pulses together, which results in more low-end torque. The downside of this extra torque production is that the engine must work harder to finally expel these exhaust pulses.

The stacking of exhaust pulses assists the rising piston on the exhaust stroke pushing out the spent gases by scavenging exhaust from the combustion chamber, which also helps pull in the new intake charge.

Therefore, while extra low-end torque is always welcome, more gain is beneficial by minimizing the pumping losses of expelling the spent gases.

## Mufflers Silencing Noise

Let us look at the dynamics of sound first. Sound is a pressure wave that forms from pulses of alternating high and low air pressure. These pulses travel through the air at the speed of sound. Funny, isn't it, I never made the connection of sound to the speed of sound.

Sound is a vibration that travels through air as a wave. The speed of sound describes how much distance such a wave travels in a certain amount of time depending on how it expresses. In dry air with a temperature of C), the speed of sound is 770 mph (1232 km/h) or 1130 ft/s (344 m/s). The speed of sound changes in different mediums like water.

In a Harley engine, pulses create when an exhaust valve opens and a burst of high-pressure gas suddenly enters the exhaust system. The higher-pressure exhaust gas molecules collide with the lower-pressure molecules already in the header pipe, causing them to stack up on each other. They in turn stack up on the molecules progressively further down the pipe, leaving an area of low pressure behind.

Therefore, the sound waves make their way down the pipe much faster than the actual gases do. When these pressure pulses reach the Harley riders ear, the vibration in the eardrum interprets them as sound by the brain. Sound wave

frequency determines the pitch of the sound. Higher wave frequencies mean that the air pressures are oscillating faster much like the high pitch whine of a Japanese bike winding out. Therefore, when hearing a lower pitch, it indicates slower fluctuations or slower wave frequencies.

This obviously affects Harley's also because as rpms increase the pitch of the exhaust pulse fluctuations increase the faster the engine runs. A pushrod-operated Harley redlines at around 5500 rpm while an overhead cam Japanese bike is at least double that, which explains the difference in sounds, both types of bikes will make near their redlines.

Lower rpms result in slower wave pulse fluctuations that give that Harley rumble we all love.

The loudness of the noise determines by the sound wave's

amplitude, which understandably vibrate more in the eardrum. Picture a rolling wave with its peaks and valleys. We can draw an imaginary line equidistant from the crests and valleys dividing them. Amplitude is the distance from this centerline to the crests and valleys. The brain registers an increase in amplitude as a higher volume noise. Therefore, the wave's amplitude determines how loud the sound is. Sound waves with greater amplitudes move our eardrums more, and we register this sensation as a higher volume.

These differing pitch waves need silencing. There are three main silencing techniques:

1. Absorption,

2. Restriction,

3. Reflection.

A muffler may use one, two, or all three systems to eliminate, reduce, or change the tone of noise.

## Cancelling Sound

It is possible to add two or more sound waves together to cancel either the sound from both or at least end up with less sound. The eardrum vibrates with the total sound, which is the sum of the differing sounds. It is like listening to a band or orchestra. The eardrum hears the totality. Now, some sections of a piece of music may dominate by one instrument such as a guitar or drum riff, which is the eardrum vibration that the brain interprets and thus hears. Other sections will merge all the different vibrations or sounds and the brain will interpret the sum total because there is only one vibration in the ear.

## Destructive Sound Interference

If two sound waves are exactly opposite or not synchronous, or in the vernacular out of sync or more properly out of phase, with one another, so that the high part of one intersects with the valley of the other, they will add up to zero. Zero vibration cannot vibrate the eardrum. Therefore, there is no vibration for the brain to interpret as sound. This is destructive interference since; the crests of one marry the valleys of the other. It is not destructive in terms of controlling exhaust noise.

## Constructive Sound Interference

If the two identical waves are synchronous, with one another, so that the crest of one intersects with the crest of the other and the valley of one overlaps the other, they will double in The two identical waves coming into sync or phase with each other where the crests and valleys overlap then they add up into another wave with the same frequency but with double sizing of valleys and crests. This is constructive interference since; the two waves complement each other and join. This is not constructive from the point-of-view of controlling exhaust noise.

Therefore, sound waves add and subtract, which is the principle that mufflers work on. The designer wants to put destructive wave interference to work as a noise controller. All three silencing methods; absorption, restriction, and reflection use a variation of a perforated tube that fits in a chamber that may have a packing material like steel wool or fiberglass. Hence, we have the term glass packs.

## Absorption Muffling

The absorption method is the simplest of the three but probably the least effective at quelling engine noise. An absorption muffler directs the exhaust gas pulses to go straight through a pipe that perforates with holes. Surrounding this pipe is a layer of glass insulation that absorbs some of the pressure pulses. A steel housing surrounds the insulation. This housing may absorb some pressure pulses if it has multilayer construction. The benefit of absorption is that this method easily lets the exhaust through.

The absorption method reduces backpressure. Absorption mufflers produce much less restriction, but do not reduce the sound level as much as conventional mufflers

Some other mufflers have a straight through design that opens up into a chamber inside the muffler that is much larger diameter than the header pipe. Remember Bernoulli's Principle one more time. When the fast traveling exhaust pulses enter the larger chamber they slow down because as Bernoulli's Principle states; the same volume must move through in the same time. There is more room so the gases slow down because more volume has more room to move in. During this slowdown period, the exhaust gases have extra time to dwell in the sound insulation thus absorbing more noise. Then the gases exit the muffler from a smaller pipe, speeding up again because of Bernoulli's Principle; the same volume must move faster to travel through in the same time because there is less room to move in.

Muffler Body Absorption Construction

Better quality mufflers construct in three layers. There are usually two thin layers of metal with a thicker and slightly

insulated layer between them. Thus, the muffler body can rid some noise by absorbing some of the pressure pulses.

## Restriction

A restrictive muffler does not require as much engineering expertise as one that muffles through reflection, and is usually the least expensive to manufacture. Thus, we find restrictive mufflers on almost all OEM exhaust systems.

## Reflection

The most sophisticated type of muffler of the three is the reflector. The muffler often utilizes absorption principles in conjunction with reflection to make an effective high-performance silencer.

Sound is really a traveling wave. As we know, when two identical frequency waves interact out of phase, they cancel each other out. All that is left is resultant low-grade heat without noise.

Simple muffler construction involves two perforated tubes within a separate expansion chamber that is part of the muffler. The exhaust gases leave the header pipe and feed into one perforated tube known as the inlet tube. The pulses bounce off the dead end of this tube and the gases reflect out the perforations into the rest of the chamber that contains the other perforated pipe known as the outlet tube that leads into the next chamber. Thousands of small pressure pulses bounce around in the main chamber, cancelling each other out to some extent in addition to the muffler's housing absorbing them. The gases reflect through the perforations of the second tube and leave that chamber to enter the next one known as the resonator chamber and then exit the pipe out of the narrower tube, usually the same or close to the diameter as

the header pipe that is known as the restrictor pipe.

Thus, we see these exhaust pulse gases speeding up, slowing down, speeding up, slowing down and finally speeding out of the narrower restrictor pipe. The reflections destructively interfere with each other cancelling themselves if identical and in phase or subtracting one from another if not identical. Reflection design attempts to preclude in-phase waves that constructively interfere, thus magnifying noise.

## Resonators

Resonators are kind of mini-mufflers and are similar to glass-packs. The main difference is that there is no sound-absorbing fiberglass or steel wool in a resonator. Their main method of silencing is not absorption but the reflective, out-of-phase wave destructive principle, whereby identical waves

cancel while others that are not the same subtract from each other.

The resonator chamber plan has a specific length and volume capacity. Mathematic design calculations determine length and volume to allow specific waves that will cancel out specific targeted frequencies of sound.

There are many sound wave frequencies producing from differing rpm levels that propel down the header pipes into the muffler(s). These mixtures of sound frequencies cannot all destructively out-of-phase cancel each other even with the assistance of perforated tubes, expansion chambers and resonator chambers. Therefore, the muffler design purpose is to destructively interfere with or cancel the noisiest waves.

## Backpressure

An important characteristic of mufflers is the amount of backpressure they produce. Mufflers with separate expansion chambers and perforated tubing create many turns and holes through which the exhaust must travel. This in turn creates muffler restriction producing a greater backpressure. Too much restriction or backpressure takes away somewhat from engine power.

EPA dictates are taking exhaust design away from simple low-backpressure absorption mufflers that do not repress horsepower to more complex reflective and especially restrictive ones that create too much backpressure in terms of some horsepower suppression.

## Turbochargers

Turbochargers introduce some backpressure to an exhaust

system, thus making it a little quieter. All of the typical scavenging rules still apply.

Common mythology is that a turbocharger has a much hotter exhaust system than a naturally aspirated Harley. Controlling the temperature of exhaust gas is by the air-fuel mixture, spark, and cam timing and not the turbocharger that mounts off the exhaust manifold.

When designing an exhaust system, turbocharged engines follow the same rules as naturally aspirated engines with the only difference is that it will require less silencing. However, one of the silencing methodologies is restriction, and a turbocharger turbine that is part of the exhaust is just that, a restriction.

## Exhaust Pipe Sizing

There are those that espouse a bigger exhaust is better. Well, this is true if the muffler holds more volume but is it true for the header pipes?

As we all know, exhaust gas is hot. A goal is to keep the gases hot throughout the exhaust system because as air-cools down it becomes denser and therefore, heavier. The engine will need to work overtime to push a cooler and heavier mass of exhaust gas out of the muffler.

Header pipes with a larger diameter than the stock 1 ¾" will allow the slowing of exhaust gasses as per Bernoulli's Principle. This, of course will give the gasses more time to cool while en route.

Furthermore, oversize diameter headers will allow exhaust pulse entropy, defining as the degree of disorder within a

system. Before this gets over-complicated, the entropy will disrupt header tuning, as exhausts pulses will lack the tendency to line up as they would in a smaller pipe.

## Header Insulation Wrapping

Coating the entire exhaust system with an insulating material such as header wrap reduces exhaust cooling. It will assist in the retention of heat with too large a header pipe. The obvious comment, I have is to reduce the pipe header diameter to the size engine it serves and then wrap the headers.

Harley-Davidson header pipe wrapping, which is a current 2008 customizing fad borrowed from hot rods and performance-racing vehicles has definite functionality.

The insulation keeps the exhaust pulses hot, lessening the

pushing work of the engine to rid them. It also encourages the exhaust pulses to line up and scavenge each other forward.

Ceramic Coating

A ceramic thermal barrier coating also encourages exhaust gas pulses to retain heat for a faster efficient exit with less engine work and to line-up for the scavenging pulling effect.

Exhaust wrap currently looks cool. It comes in different colors and is inexpensive for the look it provides.

Ceramic coating is more expensive but may become a customizing trait in the future as customizing goes full cycle.

The point is that both pipe wrapping and ceramic coating

does work but whether the effects are worth the trouble purely from efficiency and performance viewpoints on a street Harley is debatable.

## H-D Exhaust Sizing

Exhaust header pipe sizing on Harley-Davidson's should stay at 1 ¾" unless bigger cubic inches come into play.

However, exhaust pipe length, inner diameter and of course, the muffling device if any assists greatly in controlling and directing exhaust pulses. People like Jerry Branch at Branch Flowmetrics Cylinder Head Performance Services (www.branchflowmetrics.com), have done much early and continuing research on this topic including ideal exhaust pipe length, as have the pipe manufacturers. Ideal exhaust pipe length is dependent on many things like the size of engine as

well as internal engine design, and engine modifications. Length also dictates by the inner diameter of the pipe and the type of muffler.

Conventional wisdom 30 years ago dictated a Shovelhead length to be 40 inches. I went along with this in my youth but no longer do, as the pipe should be shorter for the volume of the engine.

This 40” drag pipe length will cause the power band curve to rise in the rpm range. There will be poor performance at low rpms, midrange will be so-so, and higher rpm riding will grow into the nature of the pipes and produce tolerable performance on a stock Shovelhead but will excel on one with an increase in displacement. Lower rpm riding with drag pipes will compromise by returning exhaust pulses bastardizing the virgin charge in the combustion chamber. At high rpms, the exhaust pulses are finding their way out of the pipe and into the atmosphere without re-entering the combustion chamber.

The tuning length and diameter of the drag pipe finally finds its sweet spot in the higher rpm scavenging range

Please do not go cutting pieces off your Shovel 40-inch pipes. The correct way is to cut small pieces off and each time measure the performance on a dynamometer. Keep cutting until performance starts to dip. Of course, this is laborious and expensive.

Remember there is not one single best exhaust system, as different engines require different pipes although 2-into-1 collectors reign as a category pipe. These too, vary a great deal as header pipe length, diameter, and muffler volume play crucial roles.

Let us take an 883 Evo Sportster for example with stock displacement.

Using the standard 1 ¾” header pipe used forever on all traditional Harley’s, a length of 29-inches is the best length for producing optimal horsepower. Anything close to this length will work almost as well.

If going to 2” inner diameter pipes, these would be better suited to a 1200 XL engine or bigger rather than an 883. I use this only as an example since 1 ¾” is still the ideal diameter for the 1200 XL. However, the optimal length for maximum horsepower will now be around 23-inches.

Evo Big Twin Engines require longer pipes while the Twin Cams a little longer again. However, all the stock engines run best with 1 ¾” header pipes.

My comments are generalizations and there are exceptions but other parts or designs will offset the pipe diameter.

Shorter pipes for a particular cubic inch displacement work better at higher rpms as the exhaust pulses need to exit faster. Longer pipes work better at lower rpms since the extra restriction increases low-end torque.

Everything must coordinate so larger diameter header pipes will usually work better in conjunction with larger exhaust valves. They would also receive some consideration if the engine design were running consistently at higher rpms.

Choose your mechanic and parts adviser well.

Currently, I might choose a 2” inner diameter header for say the SE CVO 110 cubic inch Twin Cam and certainly would for a 116 incher. The S&S 124 cubic inch Twin Cam might work better with a 2 ⅛” or 2 ¼” header pipe while the S&S 145 may increase ideal pipe diameter to 2 ½”. Leaving

Harley's for a bit, generally, for an engine making 250 to 350 horsepower; the ideal pipe diameter is three to 3 ½-inches. Bigger engines require up to 4-inches, which is pretty much the maximum header pipe sizing. Exhaust diameter complicates very quickly as header pipe length and muffler size and type will compromise the above diameter choices up or down. Moreover, unfortunately there is no accurate or simple calculation for optimal exhaust pipe diameter.

This is mainly due to the styling and necessary nature of an exhaust system. Bends in the piping whether for styling or to get around a frame member, transmission, to avoid a saddlebag or whatever along with temperature fluctuations, differences in muffler design make selecting pipe diameter little more than an educated guessing game.

One exhaust system that I will state is categorically better

than others is a properly tuned 2 into 1 header pipes feeding into a collector muffler of sufficient size to have at least ten times the volume of the swept volume of the cylinders. Internal muffler construction will have the 3-layer outer shell for pulse absorption, perforated tubing and sufficient volume and expansion chamber length to destructively cancel and subtract wave pulses but not at the expense of too much restriction, unless of course the design purpose is to assist and build low-end torque.

This may sound simple in the beginning but complicates exponentially. This is why I shudder when watching riders drill muffler holes, remove baffles, cut baffles and whatever else they do without a clue. This seemingly innocuous part involves more physics and design principles than most other components of a motorcycle.

## Exhaust Effect on Fuel Delivery

An example, I will repeat is that an astute technician-operator will address exhaust backpressure before tuning a carburetor because backpressure increases or decreases will have a major affect on a problem AFR at a specific rpm range but not affect another that is fine. Carburetors have overlapping jets and needles. It is best to leave fine-tuning until after other major changes.

Scavenging

A subject that I will return to repeatedly is that one might logically think that the optimal volumetric efficiency of a cylinder would be 100%. Through headwork and longer duration cams that opens the valves for a longer time beyond the theoretical engine rotation, combined with high rpm, the principles of momentum or scavenging will come into play.

Exiting exhaust gases will act as if there is a vacuum behind them and help suck in fresh intake gases at a greater rate than normal. This is sort of like a reverse supercharger. The fuel charge packs into the cylinder beyond 100% fill. It is not out of the question to obtain up to 130% fill.

Scavenging does not occur at lower rpms because the exhaust gases are not moving fast enough to act like a vacuum. Therefore, the maximum torque moves up the rpm range because the maximum cylinder fill has moved from lower rpm to higher rpm. Since all things are not equal, the modifiers to this general discussion other than the intake components listed above will be combustion chamber configuration, squish bands, spark plug location, number of plugs per cylinder, compression, engine head temperature, quality octane gas and the exhaust

Horsepower and torque have the mathematical relationship that determines if torque increases then so will

the other at a specific constant rpm. Let us look at some basic engine principles that create torque.

- Compression equals torque.

- Compression equals cylinder fill.

- Compression depends on cylinder fill.

Cylinder fill of 100% is optimal at lower rpms but can easily increase to 130% by stacking in the air and fuel via the scavenging forces of momentum as exiting exhaust gases help suck in the incoming intake charge while both valves are open during the overlap period.

With increasing cylinder fill comes increasing cranking

compression since the piston must compress a bigger volume of air and gas molecules. Of course, the burning of this increase in cylinder fill and thus cranking compression creates more heat, which expands creating more pressure to drive the pistons down on their power strokes, rotating the flywheels with more twisting force known as torque.

Scavenging and exhaust backpressure assist mightily in creating more horsepower and torque.

## Cam Overlap, Torque, Cylinder Fill, and Exhaust

Some paragraphs in this section will be review in nature but will lead into or come after related complex subject matter that will become easier to understand.

A narrow cam lobe-separation-angle will help increase cylinder fill, which increases cranking compression, which

will deliver more torque.

Correct exhaust design for a specific engine will assist in all of this by ensuring exhaust gases leave the engine in a timely fashion and will prevent spent gases from polluting the fresh intake charge. The exhaust must be a suitable one or it will work against the engine and any potential power increases. There is no one-exhaust system that is superior neither in every situation nor for every Harley-Davidson engine.

The stock exhaust certainly is not because its primary concern is to reduce emissions not allow an engine to breath freely, thus promoting efficiency and synonymous power.

Different size engines need different diameter and length exhausts. The muffler will also need to increase in volume

capacity as a street engine gains cubic inches or swept volume. Mufflers need to be larger rather than smaller.

Complicating the exhaust valve closing timing by a narrow cam lobe-separation-angle are kinetic wave pulses of exhaust gases returning up the pipes. Many think that the exiting exhaust gases simply flow out the exhaust header into the muffler and out into the atmosphere. Well they do, that is after they turn around and return up the pipe many times.

Heat or temperature differentials create negative pressures (vacuum) that are greatest near the exhaust manifold with positive pressure near the muffler. This is just a moderately fancy way of saying that it is hotter at the exhaust manifold and cooler near the end of the muffler. Heat expands and cold contracts.

The exiting exhaust gases cool as they go out the header

and muffler. This creates a negative pressure back near the exhaust port causing the cooler positive exhaust to turn around in the muffler and travel back up the exhaust reversing the flow of that particular exhaust pulse. The returning exhaust pulses must bounce of a closed exhaust valve to prevent polluting the fresh intake charge in the combustion chamber and cylinder.

Exhaust systems are perhaps the most complicated design part of a motorcycle. I hope that the pulses will bounce off a closed exhaust valve but perhaps the novice fooling with his or her exhaust will create the problem of exhaust pulses entering back into the combustion chamber past an open exhaust valve.

Tight Cam Lobe Centerlines resulting in a narrow Lobe Separation Angle increases cylinder pressure and thus torque.

The negative consequences of this are a rougher idle and the danger of exhaust reversion back into the chamber bastardizing the fresh intake charge. Tight LC's and the consequent narrow LSA will open the exhaust valve later and will close it later. The later closing creates the danger of exhaust reversion because in this scenario the intake valve also opens earlier. This creates greater valve overlap. If exhaust reversion becomes an issue, then changing the degreeing of the cam lobe centerline may be necessary as a slight adjustment to balance out potential exhaust reversion.

Once again cam design becomes art, where finding the fine line between increasing torque balances between the potential problems of exhaust reversion.

Torque cones also known as anti-reversion valves prevent reversion of exhaust gases back into the combustion chamber as they travel back up the pipes. Therefore, a torque valve preserves engine torque by preventing exhaust gas pulse

reversion back into the combustion chamber. Torque valves sometimes cast into the heads exhaust ports where they meet the exhaust header pipe. The Twin Cam TC88 head is a case in point. The exhaust port end is not round but squishes in appearance. This design rebuffs returning pulses, again breaking up or disrupting the wave pattern.

The TC96 head eliminates this feature, so the exhaust pipe will have a design feature to assist in this crucial prevention of exhaust reversion thus preserving engine torque.

For example, to understand the complexity, too much exhaust backpressure will negate horsepower torque increases from a freer flowing cam profile that will produce more power if the exhaust cooperated in the breathing department.

The Exhaust System to the Rescue

However, exhaust changes will help immensely in correcting rich conditions because the backpressure pulse waves can change the ease or difficulty of scavenging burned exhaust gases. Therefore, as a generalization I can say that an overly free flowing exhaust will usually cause a mid range dip of the air fuel curve, which is the same as saying the AFR ratio enriches too much below an acceptable lower range of 12:1. This enrichment increase translates into a drop in torque ft-lbs of power in this rpm range.

This situation exacerbates with performance cams whereby the torque drop increases without an exhaust change. Correcting this condition through carburetor jetting alone will certainly lean out the engine higher up to the point the increasing leanness (higher AFR ratio) as the engine progresses into higher ranges may even damage the engine.

The balancing act continues. The exhaust system change will do much to correct this AFR dip with carburetor tuning coming afterwards to finesse the AFR rpm versus air-fuel-ratio graph line straightening. Carburetor jetting, EFI fuel modification, ignition timing, sufficient air intake and sufficient air outtake also known as exhaust backpressure all interweave to increase or decrease power substantially.

Tuning each in isolation can afford a few extra ponies but the next tuning operation may take away from the first. Tuning must encompass all these factors in combination with each other not in isolation. Different problems require differing techniques or order of tuning.

With any motor, I favor installing a correct exhaust next after a free flowing air breather since the engine is much like the human lung where air must breathe in and breathe away without impediment.

With hemi-head, Harley-Davidson's like the Shovel, Pan, Knuckle, and Ironhead XL, exhaust tuning moves up the priority scale, as this is where the major problems first occur.

## Cam LCA, Overlap, Exhaust, and Valve Closing Delays

As I said earlier and it bears repeating that, the Lobe Centerline Angle affects valve overlap around piston top dead center. A tight LCA increases cylinder pressure and thus torque with a rougher idle and the danger of exhaust reversion back into the chamber bastardizing the fresh intake charge In addition, the LCA establishes the amount of intake or exhaust valve closure delay there is beyond the theoretical end of the relevant stroke. As the piston descends to the bottom of the cylinder but before the piston reaches the bottom, the exhaust valve begins to open.

Most of the fuel has burned. Cylinder pressure will begin to push these burnt exhaust gases out into the exhaust port and the exhaust headers into the mufflers. After the piston passes Bottom Dead Center (BDC), it begins ascending the cylinder on the exhaust stroke. This upwards action forces the remainder of the spent exhaust gases out of the cylinder and combustion chamber to make room for the fresh, clean charge of air and gas.

As the piston moves toward the top of the cylinder, the already opening exhaust valve goes through maximum lift and begins to close. Just before the piston reaches the top of its exhaust stroke (TDC), the intake valve begins to open but the exhaust valve is not yet fully closed.

Theoretically, this should not work unless other forces of physics assist like the Principle of The piston's upward exhaust stroke is removing just about all of the spent charge

by the time the piston approaches top-dead-center. As I said, just before this, the intake valve begins to open slowly. Scavenging (Principle of Momentum) begins. The remaining exhaust gases rushing out into the exhaust will draw or suck in the start of the intake charge. Conversely, the developing momentum of the intake charge also assists in pushing out the remaining exhaust gases. It becomes a mutually beneficial affair. Scavenging is how the engine flushes out all of the spent fuel charge.

Again, some of the new intake charge gases may escape into the exhaust. Thus, valve opening and closing timing is crucial.

Finding Nirvana where pollutants fully exit but the beginning of the fresh intake charge is not lost is the science and many times the art of valve timing.

Once the piston passes through the exhaust stroke ending theoretically at TDC and starts back down on the intake stroke, the intake charge pulls in quickly. Therefore, the exhaust valve must close at precisely the right point after the TDC to ensure prevention of any burnt gas re-entering.

Overlap is this area around TDC where both intake and exhaust valves are open. This is one of the most critical moments in the engine's rotational cycle. Intake and exhaust valve opening and closing must position correctly with the top-dead-center of the piston. To complicate matters the function design of the motorcycle changes this positioning for different types of riding and other factors. Different types of riding demands different overlap periods.

As the engine passes through the overlap phase, the intake valve is opening very quickly while the exhaust valve closes

just after the piston starts descending.

## Conclusion

In review, many manufactured exhaust pipes today are all about styling with no regard for performance. They are too skinny, too fat, too long or too whatever to produce good gains in horsepower. There is no one-production exhaust that I or anyone else can say is the best overall.

In general, I can probably narrow the search quite effectively to a two-into-one collector system with a bigger muffler and a restrictor pipe with the same diameter as the header pipe.

However, the consumer still needs much more information to decide on which collector system is best for his or her

application, needs, and wants. The predeing article will do much to alleviate the information void.

1 S&S Cycle, Inc. S&S Catalogue 19. High Performance Parts For American Motorcycles Since 1958. Page 8-2.S&S Super Stock Cylinder Heads and Kits for 1999-Up Big Twin Engines.

# Chapter IX

## Fuel Delivery

Please do not feel left behind when I use terms like emulsion tube or venturi in the general opening overview of how a carburetor works. I will explain these terms and others I am about to use as I progress through this chapter. I also personally found carburetion and electrical, difficult areas of learning whereas EFI for some weird reason seems easy to understand.

Perhaps, it is not so weird because it is easier. Maybe, it is because EFI is controlled and I am an admitted control freak…just kidding, well, not really…but EFI is a forced

procedure whereas carburetion depends on so many complex laws of physics to operate.

## How a Carburetor Works

Air from outside the air breather sucks in past the air filter tumultuously like an unruly river, drawn initially by the vacuum created by the descending pistons on their intake stroke.

Higher rpm riding will forcefully pull in greater quantities of air via the Principle of Momentum as scavenging exhaust gases leaving the cylinder, into the exhaust port and then the exhaust header pipes create a suction force.

The air horn, if there is one, at the entranceway of the carburetor throat extends the throat and has the effect of

smoothing and organizing this disruptive and slower moving turbulent airflow into laminar higher velocity flow. The air rushing by the emulsion tube in the carburetor venturi picks up fuel from the top portion of this tube. The main jet in the lower part of the tube controls the fuel flow volume. At full throttle this simple system works efficiently providing a constant air-fuel ratio (AFR), over a wide range of airflow rates.

At partial throttle, some types of carburetors use an intermediate jet to control the AFR mixture. Other carburetors use a tapered slide needle to perform this function. Tuning a carburetor involves selecting jet sizing and/or needle height to choose optimal AFR ratios.

It is certainly more complicated than the simplistic version I relate here as various jets and/or needles effects, overlap the others making for compromise solutions.

Poor exhaust choice is a major complicating factor for carburetor tuning. It will not be the first time I write this, but exhaust tuning known simply as picking a correct exhaust for a particular engine with adequate backpressure to control returning exhaust pulses should occur before attempting to tune the carburetor.

## How EFI Works

EFI may seem more complicated but is not. EFI evolved out of carburetion around three decades ago as meeting emissions dictates with specific AFR's for cruising, cold idle, hot idle and acceleration complicated the carburetion plumbing. Make no mistake, carburetion did meet initial, but strict EPA dictates quite successfully. Reluctantly, carburetion can still do this in 2009. It just became simpler to evolve into

EFI.

The first goal of electronic fuel injection is to measure airflow through the manifold so the computer will know how much fuel to inject. The manifold absolute pressure (MAP), sensor screws, or plugs into the manifold to measure vacuum inside. MAP sends an electrical signal via wiring to the electronic control module (ECU or ECM), computer that will send an electrical signal via wiring to the fuel injectors to open for a specific time known as pulse width measuring the precise amount of fuel to inject or squirt.

There is less dependence on laws of physics except perhaps for the measuring of manifold vacuum since an injector squirts a precise amount of fuel measured by orifice sizing the time it is on or open. Other sensors such as crank position, throttle opening, engine temperature, and ambient conditions etc. add to the sophistication and preciseness of fuel injection. Ambient is a term mechanics see thrown around all the time.

It defines as in the immediately surrounding area.

A fuel pump inside the gas tank keeps pressurized fuel at the ready all the time. Automobiles may use a bulky air-fuel meter to constantly measure conditions for real-time delivery specifications. These are too big for use with the Harley inefficient Magneti-Marelli (MM), EFI (1995-2001), and the efficient Delphi EFI (2001-present).

Carbureted bikes are much faster when comparing to early MM EFI but do not go betting pink slips against the Delphi EFI. Harley EFI gets around the use of a meter by testing airflow at all throttle, engine speed and load conditions on a dynamometer in a laboratory. The MAP sensor measures engine load as vacuum or lack of vacuum, indicating either accelerating load conditions or non-load cruising situations. From this data evolves a table also known as a fuel map that

lists or maps an individual cell for every load, throttle position, and engine speed situation. This map or table may also link to a spark map or table so that spark ignition timing and fuel delivery coordinate.

Engine sensors now deliver electrical pulse information to the ECU, which simply refers to the fuel/spark map to determine how much fuel pulse width and advance-or-retard spark timing to send to the injectors and spark plugs. The settings are nothing more than a memorized response map from the dyno laboratory.

EFI is so simple because it does all the work. However, with performance changes like cams or increasing displacement airflow will change. Simple changes like air breather and exhaust upgrades may pass with stock memorized settings because some Harley's deliver with a

slightly rich AFR from the Factory. The memorized settings will not match dramatic airflow changes causing poor performance. More importantly, there is also a risk of engine AFR injuriously leaning out. Therefore, the performance mechanic must change the ECU memory; modify input to the computer ECU memory or modify computer-memory-output signals.

Ignition changes are not a big concern with the Twin Cam but are necessary with Evolution Big Twins. There will be more on this later in subsection: EFI Ignition and Fuel Modifiers.

Carburetor Performancing

Air breather, exhaust, cams, and increasing displacement have different fuel and spark timing requirements. These are separate components on a carbureted Harley so there are two

areas that need adjustment to accommodate performance changes.

I will discuss performance ignition as I progress.

Carburetion changes may be as simple as jetting/needle changes or complicate further with bigger venturi requirements and all that goes with structural changes to the carburetor itself.

There are compelling arguments to change the type of carburetion entirely on pre-1989 Big Twins, as they are all the inefficient butterfly plate-opening types that let differing graduated amounts of air through the venturi. The problem is that it is a crude system and the butterfly plate disrupts the laminar flow velocity of the air-fuel mixture.

In my opinion, the S&S Super E or G carburetors (www.sscycle.com) are some of the best of these butterfly type carburetors provided the engine has large enough displacement to accommodate them. These carburetors are simple, reliable and deliver lots of gas for almost any size big engine. When I use them, I like to add another fuel circuit by installing a Thunder Jet (www.thunderproducts.com), that resolves any mid-range lean situations as well as adding fuel to the normally starving top end especially with stroked engines.

Twin Cams and 1989 to 1999 Evolutions use a slide/ butterfly type carburetor made by a subsidiary of Honda called the CV Kehein. Do not confuse the CV Kehein with its predecessor, the Kehein butterfly-type-carburetor in use from 1976 to 1988 on the Big Twins. The CV Kehein is the all-time best carburetor ever used on a Harley-Davidson. It lends itself well to performancing.

The slide Mikuni Smoothbore is probably the only carburetor I ever entertain using over the CV Kehein. In my opinion, the Mikuni Smoothbore is marginally superior to the CV Kehein and light years ahead of the Kehein, Bendix, Tillotson, and Linkert butterfly type carburetors used back four or more decades.

There are many other performance carburetors such as the Big Air Carburetors, (www.mcadvantages.com). This carburetor will fit from the small 74 cubic inches like the 1200cc Sportster to monster 155 cubic inch engines. They claim new technology with highly atomized fuel vaporization that keeps fuel in suspension longer for better throttle response. There are three external fuel knobs for adjustment. They make a claim of a 15-horsepower increase. I doubt the same horsepower increase will be constant over such a wide engine range since 15 horsepower is exceptional for a 1200 cc Sportster but certainly not as impressive, percentage-wise,

with a 155 cubic inch behemoth. Incredibly, the Big Air Carburetor meets the brutal 2010 EPA regulations.

However, it is difficult to beat the Smoothbore in terms of simplicity, performance, and reliability although monster engines may dictate the use of other carburetion.

Performancing EFI

As I have said, the ECU internal map or table settings that also link to a spark map or table to enable coordination of spark ignition and fuel delivery are a memorized response map from the dyno laboratory and not real-time diagnostic responses. The sensors that are part of this engine management system deliver electrical pulse information to the ECU, which simply refers to the fuel/spark map to determine how much fuel pulse width and spark timing to send to the injectors and spark plugs. It bears repeating that the

memorized settings are for stock emissions control air breathers, induction sizing, cams, heads, piston displacement, and exhaust. These settings may accommodate an air breather and/or exhaust change since many times the Factory motorcycles meet the EPA regulations under the mandate. In these cases, Harley will richen the settings up to the mandated allowable settings. Therefore, conceivably the AFR often allows these simple breather-exhaust changes to grow into the slightly richer condition.

Do not take this statement to mean that it will always happen!

Always check AFR after making any changes as excessive leanness and inducing heat may injure the engine.

However, once the performance mechanic goes beyond

simple changes and broaches other more complex ones such as a cam change then certainly the memorized fuel map settings will be wrong since airflow changes. The airflow changes invariable mean an increase in airflow that will lean out the bike without the addition of more fuel. I will delve into this subject in detail in a following section, Ignition and Fuel Modifiers.

## Ethanol Fuel

Ethanol fuel may seem less expensive but makes less heat energy. Heat energy measures in British Thermal Units (BTU). More ethanol will need to burn to produce equal energy to the burning of a smaller amount of gasoline.

It will not hurt your engine to burn a tank of ethanol if that is all that is available as long as it is used quickly…like don't leave it in your tank for a week for a few reasons.

- Ethanol is corrosive to sheet metal.

- Ethanol requires the use of stainless steel gas tanks.

- It will also peel off tank sealer inside the tank, which is a nightmare to repair as the sediment constantly bungs up fuel filters and smaller orifices.

- Ethanol attacks rubber gas lines and destroys their usefulness.

- High concentrations of ethanol are caustic attacking the aluminum in EFI induction throats and carburetor bodies.

- The rubber used for sealing manifolds and the O-rings

found inside carburetors will expand and just go wonky leaving fuel to leak and excess air to enter and lean out the engine.

- All rubber needs upgrading to Viton-based manufactured seals, O-rings and fuel line.

Read your H-D manual to determine the allowable percentage amount of ethanol in gasoline for year and model Harley-Davidson.

However, ethanol is a clean Octane.

Food shortages, particularly in the third world and emerging economies are in conflict with incomes that are more lucrative for farmers growing the grains for ethanol production. Defenders say that rising energy prices and

fertilizer costs are more of a factor with rising food costs with ethanol only contributing fifteen per cent to food costs. In fact, some calculate that only 15% of the rise in food costs is due to making ethanol. Furthermore, others feel that lobbyists in the oil industry are spreading false rumors and misinformation about ethanol to defeat its usage in gasoline. These ethanol supporters say that the protein and other nutrients are mere byproducts; that only the starch in the plant fiber is useful for ethanol production.

Still other defenders of ethanol point out that this less expensive product keeps the more expensive oil-based gasoline in check; that gas would cost much more without the inclusion of ethanol.

## Octane

Octane prevents gas explosions and encourages burning at

a defined rate.

One of the great advances in the early 1900's of engine technology was the discovery of Octane. There are many different Octane substances or elements, the best early example is lead, which control the rate of burn.

Before Octane, haphazard-detonating explosions destructively caused the first engines to hack along with much inefficiency. Octane has the ability to stop chaotic explosions and bring order to the combustion chamber. Octane is a very specific compound comprising of differing chemicals in use for comparative testing of detonation resistance of other fuels.

Gasoline is a complex mixture of many chemicals. Chemical components add and delete to enhance the Octane rating. Tetraethyl Lead was the old and easy way; now it is a

lot more complex. Lead an excellent Octane is also corrosive to the internal engine parts including spark plugs and exhausts. Some Octane's are oxygenates (MTBE and all the alcohols) that also lean the mixture and diminish power unless we compensate for them.

Enhanced Octane reduces detonation harm inside the engine but many times produces harm to the environment and to us. Lead, for example causes mental retardation and learning disabilities in children. It does so to a lesser extent with adults. The reason is that children will put substances in their mouths at one time or another that adults will not. A very dangerous quality is that lead tastes good. In the day, engine exhaust left lead residue everywhere on the ground and in the soil where children play.

Successive Octane's did not wreak havoc with children's

brains but they did cause cancer with all that exhaust permeating the air with silent debilitating and sometimes-carcinogenic killing fumes.

The advantage of the alcohols like ethanol is they burn cleanly. They are also a safe Octane enhancement.

However, Harley-Davidson needs to use Viton resistant-rubber compounds in manifold seals and other induction components as ethanol will distort traditionally used rubber composites. Harley-Davidson warns against high ethanol levels in pump gasoline for this reason. Consult your manual for the acceptable ethanol level for your year and model. Gas pump signage usually has percentage levels of contained ethanol near the price per gallon notice.

An Octane enhancer discourages unwarranted and

unwanted explosions. Octane causes fuel to burn at a predetermined rate. How long it takes a flame front to travel across a combustion chamber with controlling Octane allows the engine designer and tuner alike to advance ignition timing or retard it. The spark can ignite the fuel at the most auspicious time for maximum fuel burning known as the percentage of burn, thus creating maximum controlled heat.

Rate of burn, piston speed, and cylinder diameter are some of the main factors determining ignition timing. Cam timing, when valves open and close, will also affect ignition timing. Thus, the complex blueprint of valves opening and closing on previous and succeeding engine strokes and the spark ignition occurring before a compressing piston reaches the top of its stroke is all about combustion lag.

Combustion lag is the time it takes to get gas burning, the rate of burn, sizing of cylinders, mechanical parts movement lag time, piston speed and combustion lag all determine

ignition timing.

## Air Filtration and Power

Air cleaners and the inside filtering material are one of the oft neglected but crucial components of the performance equation. Induction is all about the delivery of air for both an efficient running engine and the performance one.

The basic tenet of power is air. Coincidentally, the same applies to engine efficiency. More power potential will result from the ingestion of more air.

Gas cannot burn unless mixed well in differing proportions depending on need such as acceleration with the oxygen in the air. We cannot burn more gas without more air in correct proportions. Therefore, if we want more power the volume of

air intake must increase.

## Velocity Stacks

A dragster that is rebuilt every few passes need not concern itself with fine dirt-borne air. Drag bikes use long velocity stacks without filtration, as the goal is to get as much air as possible into the engine. There is no such thing as too much air for these bikes because their mega cubic inch displacements easily accommodate the volume. The last statement is a bit of an exaggeration because air delivery measured in cubic feet per minute (cfm), should closely match the cylinder capacity.

Proponents of the velocity stack view an air filter as an impediment to gaining maximum power, which is dependent on maximum air. A velocity stack attaches to the entryway of the carburetor throat or the fuel injection throttle body and

fundamentally extends the induction system. The stack provides a smooth, radiused entranceway for air to flow into seamlessly. This seemingly innocuous statement has huge ramifications for increasing air delivery.

The radius at the mouth of the stack functions as a transition system for turbulent air to smooth out. This increases the potential for more delivery volume. Turbulent air moves in every which direction decreasing volume of flow. The stack as its full name denotes increases air velocity by ensuring smoother airflow. The velocity stack increases power because of the delivery of more air to mix with the fuel.

Velocity stacks may show great power gains on a dyno with the bike at rest because there is no air turbulence with which to contend. However, with air turbulence at speed round the stack's mouth, there may be quite different results

especially in certain rpm ranges. The bike may cough and spit back for a bit, then as airflow smoothes out, take off like a bullet.

The radiused mouth blueprint, initial stack diameter, the taper of the inner diameter and the length of the stack for the size engine served control these dynamics and are crucial for air delivery gains.

An air horn is the curvature of the radiused mouth of the stack. The air horn causes turbulent air rushing by it to smooth out. At first glance, a simple induction extension rapidly becomes a study in complex physics best left to the likes of the professionals like Lee Wimmer (www.wimmermachine.com), of Wimmer Custom Cycles or the venerable S&S Cycle, Inc. (www.sscycle.com), or Tim Doherty (www.dohertymachine.com), of Doherty Machine or Bob Wood (www.woodcarbs.com), of Wood Performance Carburetors and other professionals in the industry.

Velocity stacks are normally unfiltered and therefore contribute to premature engine wear because of delivering unfiltered impurities through the induction system and into the engine.

## Clean Air

Street bikes, whether performanced or not, have an expectation of longevity before a rebuild. Longevity to a conservative touring rider on a stock machine may well expect in excess of a hundred thousand miles before a major teardown. The same expectation for mileage will be a lot less for a mega-cubic inch street bike that tears up the urban asphalt. Engine life is relative to the machine and its use.

The point is that unfiltered air reduces longevity. Some

particulate in the air may well burn in the combustion process preventing any harm. However, it is a guarantee that not all of it will. Some will adhere on cylinder walls forming an abrasive for the rings to push and pull up and down the cylinders. Other unwelcome particulate may also settle on the cylinder bore below the ring lands wreaking havoc on wearing piston rings, piston skirt thrust surfaces and the cylinder sleeve itself. Cylinder and piston material is relatively soft compared with other metals in the engine so these areas are particularly vulnerable. A smaller amount will blow-by the rings into the bottom end. Other particles may get between the valve and the guide. Dirt mixed with lubricating oil will form an abrasive that wears parts prematurely.

Have I talked the reader into a superior air filter yet?

Proportionate Air

On a street bike, there is a potential problem with too much air for a particular size engine. Therefore, it is possible to have too big an induction system for some size engines although with Harley-Davidson this is not a common problem.

The indicator of a performance filter performing is a resultant need to up jet the carburetor to allow more gas flow because of too much air relative to the supplied gas known as a lean condition.

Electronic fuel injection most times adjusts fuel delivery to air intake automatically via sensor information delivered to the computer ECM but may need the pulse width increased, which is the equivalent of up jetting a carburetor. Pulse width is the length of time the injectors squirt gas. This is a more practical alternative with minor power increases than actually replacing the injectors with larger ones.

I talk about form over function all the time. This occurs when a style or look dominates and eclipses the functionality of the part. In the wacky world of Harley-Davidson, the conflict between perceived good looks, good sound, good whatever, and how well something performs many times is like the biblical fight between good and evil. Evil wins out when some trick looking air cleaner makes its way onto a Harley that does not flow enough air for a mini-bike. This will suppress existing horsepower.

We also must deal with other power suppressions like restrictive emissions controls. Speaking of which, all later Evolutions and Twin Cams have progressively more suppressed power just waiting to be unleashed. The newer the bike, the more suppressed power occurs. Emissions controls,

which include the cams, exhaust, ignition, and fuel induction including the air cleaner and filter, repress power for the single-minded goal of meeting EPA standards in reducing emissions.

Evil will also win if the ultimate fashion air cleaner's filter is such poor quality and design that it lets lots of air through carrying lots of dirt. We need to filter the air entering the engine from a practicality point of view.

The filter design for a performance street bike need not impede air. We can have the best of both worlds. I will examine some air cleaners that marry both form and function for great looks and good performance gains.

Crucial Filter Spacing

It is important that at least a quarter-inch of space exist

between filter and the outer cover. I do not want to restrict airflow by scrunching the filter between housing surfaces. I cannot tell you the number of stock Harley's over the years that required jamming the outer cover up against the filter and then tightening the cover further against it.

The Environmental Protection Agency imposes two more duties on top of filtration and cfm delivery on stock Harley-Davidson filters. The first is meeting emissions controls in terms of pollutants. The second is to dampen air manifold noise; hence, no spacing between filter and housing for noise to travel around.

A stock filter is not the performance enthusiast's friend.

## Cleaning Air Filters

Consult your manual for the maintenance schedule for particular year and model. If using an aftermarket filter follow the instructions given by that manufacturer.

## Maintenance Schedule

However, it is prudent to service the air filter after 1000 miles (1600 km), and every 5000 miles (8000 km), thereafter in normal riding Do so more often if riding in the rain frequently. Dirt roads or dusty riding conditions require more numerous inspections, cleanings, and/or replacements. Symptoms of dirty filters are hard starting, anemic performance, and poor gas mileage. Yes, there are other conditions that may exhibit the same symptoms, but I always proceed with easy fixes that are inexpensive and progress to the more complicated and expensive ones. This is so obvious and logical.

The nature of many Harley riders and mechanics that should know better is to bypass the obvious and diagnose the most expensive, time-consuming fix right out of the gate. Many times, after much money is spent and time consumed, both mechanic and rider realize the cause is something simpler. It takes minimal time and effort to eliminate the air filter as the cause of the above noted symptoms. A dirty filter may restrict air enough to cause a rich condition that will use more gas and can ultimately foul spark plugs.

Increased gas relative to air feeds into the combustion chamber. This decreases mileage because of inefficient burning of the fuel. Excess gas discards out the exhaust as unburned or partially burned pollutants. If the condition becomes too rich, the excess gas may foul the spark plug(s) with a flat black fluffy residue preventing the spark from arcing between plug electrodes.

As I said, a clogged filter restricting air-intake might also cause hard starting. After checking the filter, we can go onto progressively more complicated diagnosis for hard starting. For example on a bike with electronic fuel injection, check the idle air control (IAC), for incorrect positioning at rest. If the IAC positions wrongly, it will send the wrong information to the electronic ignition module (ECM). This easily happens if it gets sticky and does not set to the position dictated by the ECM. Incorrect IAC position will result in the wrong amount of air delivery at closed throttle and one symptom may be hard starting. Another cause may be disturbed or leaky manifold rubbers, again upsetting the air balance resulting in a symptom of a hard starting. I mention all this to show how complicated some people can make a simple problem in bypassing a potentially dirty air filter.

How can performance be what is expected with restricted air-intake? Lastly, how can gas burn at a high percentage rate

when the catalyst for ignition, air is lacking?

Many times, the simple maintenance, so important to a functioning motorcycle and crucial to a high performance one is lacking. I watch so many spend big money on super-duper engines that fail to ensure clean filtration, proper tire pressures, belt and chain adjustments and on it goes. More than once, I beat performanced bikes not maintained correctly, in impromptu races with a stock but well cared for Harley.

A simple but effective trick if the bike is equipped with a round air filter is a scheduled rotation procedure. One section will get dirty faster than the rest of the filter, as there will be a sweet spot for more air intake. Rotating a round filter will extend its life and keep the whole filter cleaner for a longer time-period.

Air Filter Materials

Tim Doherty of Doherty Machine assumes that anyone interested in an efficient running engine never mind a performanced one will use the best and widest pleated filter available that both filters the air and allows sufficient volume to supply adequately the size engine served. For Tim, this is the easy part. He concerns himself with the difficult physics of controlling the volume of air once past the filtration stage.

Tim explains "…in reference to flowing more air and I make this reference with strong and proven conviction, an air cleaner is simply what the name implies, it's an air cleaner and any given engine will benefit by installing an air filter capable of supplying a specific engine's air requirements. The key to this entire matter is realizing that and an air filter is simply that a Removing the stock restricted filter and using a bath towel could in simple terms accomplish air mass and the filtering process. The key to air filtration is controlling the air once it passes the filtering stage." 1

Thus, the air cleaner has three functions,

- Filter airborne dirt particulate,
- Allow sufficient cfm volume through the filter,
- Control the airflow stream after filtration.

The use of four basic products, paper, foam, synthetic media, and natural cotton gauze formulate most air filters. Pleating paper, synthetic media, and gauze is a common design feature used to gain more surface area from which the air can penetrate and pass through. All three have a myriad of different size holes in the filter material to trap particles of dirt. Pleating offers another advantage with increasing the number of entrapment holes. More holes will be available with more filter surface area. A single paper or gauze layer

will trap dirt particles that are larger than the hole they attempt to enter. A problem arises when the particle of dirt is smaller than the hole as it will obviously get through.

Once a hole captures a dirt particle it plugs. Fortunately, there are a great many holes. As a high percentage of the holes plug, the now dirty filter becomes restrictive and does not allow enough air in. Hence, the needs to either wash or change filters.

Paper does not like high humidity moisture and likes rain even less. Moisture will swell paper. This restricts air traveling through the filter. Furthermore, the sucking force of engine vacuum on a moistened paper filter element may cause embedded, captured dirt to further clog when the paper dries out.

Have I talked the reader out of paper filters yet?

## Paper Filter Elements

Actually, paper is not that bad for smaller workhorse engines that are not performance-oriented. Some good filters for these type engines use paper although my personal preferences are multi-layered cotton gauze or synthetic media. Paper usually has a single layer while gauze has multiple interwoven layers. Paper filters have relatively small filtering holes. This is necessary to capture a higher percentage of particles because smaller specks of dirt can pass unhindered through larger filtration holes. Therefore, cfm air delivery is restricted for larger demands. Paper filters surface load dirt. The surface of the paper filter traps and captures particulate.

Reputable and knowledgeable people like Tim Doherty feel that the type of material is not that relevant to performance

increases as long as the cfm rating is the same requirement for the size engine served. Of course, Tim's high flow filters use the best of filtration-pleated materials, which does not include paper.

In my opinion, adequate paper filtration is incapable of high cfm ratings required on many of today's monster engines.

However, the perfect marriage is the necessary ability to remove particulate from the air along with substantial cubic feet per minute (cfm), air delivery.

Another factor I will discuss many times is the surface area of the filter. How can a pleated one-inch wide filter compare with say, a Doherty filter that is pleated but 2 ¾" wide?

Do not use a thin filter on a performance engine, as it will

restrict cfm delivery.

Occasionally, a paper filter may be oil impregnated from blow-by gases not properly separated by the breather apparatus in each head. Oil is supposed to separate internally, inside the engine from venting gaseous air but this is not always a perfect world. If there is oil in the paper element, discard it and install a new filter.

The Air Filter Light Test

To check a paper filter like those used in Harley-Davidson Twin Cams, hold it up to the sun or a bright light. Look for a visible, consistent, and uniform permeation of light through the filter. If not, the filter needs cleaning. Unlike most paper filters, the H-D one is cleanable. Do not wash a paper filter element in solvent. Gently wash in warm soapy water. Let it soak for around a half-hour to remove soot and particulate.

The filter element may be air-dried. If using compressed air to dry the filter: gentle and low pressure is the key. In order to remove dirt particles, blow air from the inside out. If blowing from the outside, there is a risk of further embedding dirt particulate.

Do not put oil on a paper filter to assist as a filtering capture agent.

Do the sun test once more. If the filter fails a uniform sun penetration test, discard it and install a new one.

If a new purchase is required why not upgrade to a superior flowing filter, which will have more surface area and in all likelihood be of a natural or synthetic media material.

Foam Filter Elements

Foam, natural cotton, and synthetic media air filters use oil as a capture agent. Therefore, if minor engine blow-by occurs it will actually assist in filtration.

Foam filters can be very superior to paper filters or they may be more restrictive especially in performance applications where engine vacuum can suck both types inward. It is important to purchase filters from reputable shops and manufacturers to avoid a restrictive airflow foam filter.

In theory, a foam filter should and usually will filter better than a single layer paper one. The following explanation will demonstrate the superiority of a properly made foam filter over a paper one. Paper filtering holes are a straight run for air-laden particles where luck of the draw dictates a smaller

particle will pass through a larger hole. Foam filters actually have very large holes but capture particulate in a different way. If the larger holes were a straight passage, the foam filter becomes useless. The foam filter holes are not straight at all but wind their way through the foam in many directions. The larger hole-design is to ensure all the dirt is able to enter. As dirt, traveling in the air hits a wall when the tunnel hole changes direction; the impregnated oiled filter will grab onto the dirt and hold it. Since the holes are large, more dirt can enter and the already captured particulate, soaked in oil will assist in the new capture. If dirt makes it, around one or more bends another is likely to capture it. It is difficult for the dirt to make it all the way through especially with the sticky oil-filtering agent. The paper element holes capture one particle each and clog. The foam filter holes capture many particles before clogging becomes an issue.

Furthermore, the foam filter is reusable after washing in solvent and re-oiled according to manufacturer specifications.

## Natural and Synthetic Media Elements

Like paper elements, the filtering holes in natural and synthetic media elements are mostly straight through and of different sizing. However, the comparison between paper and media ends with this similarity.

The cotton gauze used by companies such as K&N can easily flow two or three times the air than paper can. This is a generalized blanket statement designed to demonstrate the superiority of layered cotton gauze over paper. There are other parameters like how big the comparative filters are as well as the number and depth of pleats. This dramatically affects surface area that will determine the actual percentage superiority.

In order to filter air adequately, paper must be thick or conversely densely compressed otherwise particulate will easily traverse the material. Unfortunately, this makes paper more restrictive to airflow. Then the paper progressively clogs as it collects particulate. A pressure differential begins to occur. Pressure drops inside the filter whilst remaining at atmospheric pressure on the outside. In extreme cases, dirt will forcefully suck through the filter and into the engine. This demonstrates the importance of having a clean paper air filter at all times.

Natural filter elements are usually cotton gauze layers overlaying each other. The overlays assist in capturing more particulate because the holes of successive layers do not line up with each other. The same situation occurs with the random sizing of the holes.

Once again, filter oil acts as an effective capture agent

K&N (www.knpowersports.com), perhaps the best-known supplier of filters uses a 4-ply natural cotton gauze material. The advantage is the very small natural filtering fibers that make up cotton gauze that traditional paper and foam have difficulty in replicating. Layered-gauze does not surface-load trapped dirt like paper. Instead, the natural cotton gauze and some synthetic media filters exhibit depth-loading characteristics where retention of filtered dirt particles is multi-level throughout the different layering.

## Capturing Particulate in a Filter

The theory of capturing particulate gets a little more complicated as we look at the characteristics of flowing air and its ability to carry dust and dirt.

## Interception Filtering

Interception is the first concept. Airflow like water will always take the shortest, fastest, and most direct pathway. As the airflow travels around the small cotton fibers seeking passageways through the filter, some of the airborne particulate will make contact with the fibers. The oil-impregnated fibers will arrest and capture these particles. There may also be a tacking material in some synthetic media filters fulfilling the same capture function as oil.

## Impaction (Collision) Filtration

Unlike smaller, lighter airborne dirt that unequivocally follows in the airflow, larger, heavy particulate have their own physical properties where inertia and momentum cause them to deviate from turning air flow. The concept of impaction or collision comes into play. The larger particles

momentum forces them to deviate from the air stream as the air turns seeking the shortest path of least resistance through the cotton gauze. As the air turns, the bigger particles keep going straight and collide head-on with the capturing oiled fibers.

Dust Filtration

The smallest particle dust will behave differently from small and large particulate. Within an air stream, there may be velocity differences; changes of direction and of course, pressure adjustments. Dust lacks momentum and inertial qualities to combat or deviate from these forces. The dust particles will appear to have frenzied and haphazard behavior going in complete disorder within the air stream. They too will collide with the fiber media whose oily capturing agent will collect them. This effectively captures particulate much

smaller than the larger pathway holes through the media.

Cleaning natural cotton fiber filters is easy and they are reusable. Do not use compressed air to clean them as filter material can blow out of the surrounding wire mesh, thereby ruining the filter. Also, do not scrub the wire mesh with a brush to remove sediment as the cotton gauze tears easily. These filters are washable, may be re-oiled, and are reusable like foam ones. These filters should dry naturally

Also unaffected by moisture, gauze filters may be assisted by minor oil blow-by from engine breathing.

## Performance Filters and Air Breathers

If purchasing an upgraded air filter from a respected manufacturer like K&N, one might logically assume it passes more air and therefore be a performance filter. This is often

not true, as manufacturers will generally try to have an air flow-rate that closely approximates the stock application. Therefore, a replacement filter made specifically for a particular motorcycle may not flow more air.

If the filter purchased is an adaptable one for many applications, it may well flow more air because there is no one set flow rate to emulate…or it may flow less. In my opinion, they do this so as not to cause unforeseen problems for the purchaser. Higher airflow rate filters may require more gas delivery to prevent a lean condition. From a performance perspective, this is exactly what is desirable.

It is true that an upgraded filter that flows close to the stock one may collect more air borne dirt thus making the purchase very worthwhile from a maintenance point of view. Reading the labels or talking to a competent parts person or

mechanic will lead the purchaser to the right choice for what they want.

The manufacturer will usually indicate that a filter is a performance one if it allows more airflow than a stock one for a particular application. Of course, how the airflow is calculated becomes a mitigating factor if different measurement techniques are used. I need to compare apples with apples not apples with oranges.

## The Flow Bench

I went over flow bench concepts in detail in the last chapter, Chapter VIII: Eliminating the Performance Obstructions but need to do so again because of related subject matter. If you, the reader is familiar and understands these concepts, you may want to skip this section although I present some extra information here.

The concept of airflow is difficult because it is invisible. What cannot be seen becomes complex. Therefore, many cannot understand and grasp the simple complexities of both.

We need to measure airflow. We need to know when we make performance and/or efficiency changes if airflow increases or decreases.

A flow bench measures cfm airflow. A flow bench is one of those tools like a Spintron Dyno or a chassis dynamometer that allows the performance mechanic to move ahead in the right direction on a scientific basis. Increased air is equivalent to more power.

Briefly, the function of a flow bench is a testing measurement tool. It will become readily apparent that

comparing flow rates of say, an air filter can become very complicated for the consumer almost immediately.

As Ann Christoffer, a Test Technician at S&S Cycle, Inc. who is a very proficient flow bench operator says, “As for why there are differences in airflow number’s, there are many variables, in most cases, from one bench to another, people have to take the time to make sure their bench is calibrated properly. One could easily see around 5 or more cfm difference from different benches. Operator errors, consistency in testing are other factors to consider as well as leakage from fixturing when testing. Also, testing needs to take into account barometric pressure….” 2

Air density varies with both altitude and the weather conditions. Air is denser when cool. There is more air in the same amount of space in colder weather than hot. It follows

then that your Harley will be faster in colder weather since there is more oxygen in the denser air mass. This is why an equalization process of barometric pressure must occur when comparing ratings for a filter or else effective and honest comparisons will be void.

In my opinion, as with measuring head-port flow, pinpoint accuracy in comparisons is not possible but it is easy to compare accurate flow rates from one bench to the other if all testing parameters are the same. We can compare if using among other things the same attachment to affix the head or filter, as the case may be, to the same model flow bench and use the same pressure in the test.

This is where the consumer gets into trouble and may not make the right decision in choosing a particular product. Competing products like an air cleaner and filter are usually tested on different flow benches, may not have the same affixing attachment and may be tested with different pressures

producing wildly different and conflicting results that are not comparable at all. Using other supposedly comparative tests but with different parameters skew the results out of control making one product falsely superior or inferior to another.

However, if testing the products within specific parameters they will be roughly accurate and thus produce valid comparisons.

Using cylinder head flow comparisons as an example, the air inlet on the flow bench should be radiused with all head testing. If the inlet is sharp, airflow disrupts and can easily vary with up to 25% less flow when compared with say, the same head using a radiused inlet. Correct radiusing will ensure maximum laminar airflow. Laminar defined is a smooth layered flow of a gas, in this case air, in which neighboring layers do not mix, and flow at different velocities.

## Head Flow

Head flow closely relates to but is separate from air filter flow. One feeds the other so the same principles apply. Flow benches relate to head flow in most riders minds. Heads are more complicated to test than a filter and housing especially when comparing flow rate figures. Confusing the already mentioned validating parameters is how far the valve opens for various flow tests. Testing at different valve-lift openings complicates comparisons especially when not listed in an advertisement.

Many might think that airflow at maximum cam lift when the valve is open the furthest is the appropriate comparative position. However, maximum cylinder fill will obviously occur as the valve is closing.

Also known as volumetric efficiency, we will want maximum cylinder fill to surpass one hundred percent. The performance and efficiency goal is to pack the air in up to 130 percent. This is achievable.

Again, volumetric efficiency will occur just as the valve is closing. In almost all cases, published tests will display the best result. Unfortunately, the next advertised test might show a result using a different valve lift opening. Who is to know, which head flows the most air? It is imperative to know that all testing parameters are the same or else the figures will be useless.

Even in the case of an air filter breather, one will need to know if the air passing into the induction tract is laminar or turbulent. Laminar air, flowing straight and even will get a higher rating every time. Again, a radiused inlet will allow

more airflow as turbulence converts into laminar flow.

Identification of all the flow characteristics of a filter within its breather compartment will allow changes that optimize the performance enhancement of the complete unit. Modifications to first get the most air through the filter and then quell the turbulence into a smooth laminar flow will improve the performance potential.

Test Pressure

Flow benches compare two air pressures,

1. One is a calibrated pressure across a known orifice or opening,

2. The second is a constant test pressure fed through a head, carburetor or in our case a breather.

The air feeds via an air pump much like that of a household vacuum cleaner. This continuous and constant test pressure drops across the opening. Calculating the ratio of the pressure drop relative to the calibrated known pressure determines volumetric flow, which is fancy lingo for the amount of air measured in cubic feet per minute, (cfm).

Different testing facilities and manufacturers use different test pressures making comparisons difficult for the consumer. A column of water called a manometer indicates and validates the test pressure chosen. The height of the water in the manometer column visualizes the invisible test pressure. The suction of the flow bench drawing air through the test item such as an air filter also pulls on the water in the manometer's column. The height of the water determines and reflects the air test pressure. Generally, use of between about one-and-a-

half inches and twenty-eight inches of water are typical test pressure choices. K&N and Bobby Wood of Wood Performance Carburetors use a test pressure of 1.5” H20, while Lee Wimmer of Wimmer Custom Cycle uses 10” H20 and S&S Cycle, Inc. uses 28” H20. A mathematical formula presented in the last chapter will convert and equalize the different manometer test pressures.

Measurement of actual airflow is via a second manometer measuring the pressure drop. If there was a standardized manometer test pressure that different manufacturers and technicians could agree on, the test results, everything else being equal, become comparable. Complicating matters is the use of differing manometer test pressures.

It is imperative to maintain the same test pressure as indicated in the manometer inches of water during the entire test procedure or results will again skew and become useless.

## The Law on Warranty Concerns

Can a manufacturer void their motorcycle warranty unless their air filter is used? Furthermore, does an authorized dealer have to install the air filter for warranty reasons? As with oil and oil filters, this is a common scare tactic, when used to force customer loyalty.

The United States passed the Magnuson-Moss Warranty Act in 1975 whose statute governs warranties on consumer products. Named after its sponsors, United States Senator Magnuson and Senator Moss, most other western countries have followed suit with variations of this consumer protection law. According to the Report of the House of Representatives, which accompanied the law, Congress enacted the Magnuson-Moss Act in response to the widespread misuse by merchants

of express warranties and disclaimers. The legislative history indicates that the purpose of the Act was to make warranties on consumer products more readily understood and enforceable and to provide the Federal Trade Commission with means to improve protection of consumers. 3

There is no law to force manufacturers to give a warranty on consumer products. However, if a warranty is given, which Harley-Davidson obviously does, it must comply with the Magnuson-Moss Warranty Act. This law states in part that it is against the law for a vehicle manufacturer to mandate the use of a specific air filter, oil filter or oil unless it provides these items free of

The manufacturer, however, can insist on adhering to their specifications with aftermarket replacements of filters and/or oil, which is fair. When I first started wrenching in the late

sixties, I think replacement quality was a concern. There were filters, whether it was an air or an oil filter, that were substandard and the same concern applied to oil.

In my opinion, brand name air/oil filters and oil are all of superb quality these days. A few years ago at American Iron Magazine, we tested numerous oils, both fossil and synthetic to find out the best one for a Harley-Davidson. They were all excellent including the H-D proprietary oils. Some were better than others were but only by small degrees, any would do your bike proud and none tested would cause harm.

The performance filters discussed in this article as well as a host of others on the market are of the highest quality in filtering capacity and the ability to flow high velocity and therefore high volume air.

Performance Air Breather Design

The entryway of the air cleaner to the mouth of the carburetor/EFI throat is crucial for performance. I will often refer to this as part of the induction system. This area needs a radius shaped contour. If an air breather lacks a radiused entryway that buttresses to the induction throat, then potential performance increases will suffer and engine efficiency compromised. Airflow will be turbulent entering the induction throat preventing maximum flow and resultant efficiencies.

Look at this radius contour as a mini-velocity-stack whose function is to smooth airflow going into the entryway of the carburetor throat or EFI induction module throat. Smooth in-line airflow has higher velocity and therefore, higher volumes than turbulent uncontrolled flow.

# The Doherty Power PACC Air Cleaner

Tim Doherty of Doherty Machine is a person that knows a lot about air filters and breathers. Tim has some of the best on the market called the Power PACC series. He does not feel that the filter itself is rocket science but a necessity to filter the air efficiently along with the required flow of enough cubic feet of air to supply the engine at hand. Tim also thinks that an air filter need not be an obstruction to getting damn good performance.

As he says, “The key to air filtration is controlling the air once it passes the filtering stage. I designed the Power PACC in 1999 based on the above-mentioned theory, which was to improve the airflow dynamics between the air filter element and the entrance to the carburetor or throttle body. I spent weeks on a flow bench and Dyno generating the shape and

length of the venturi used with the Power PACC. Our venturi increases the overall intake tract length 1.5 inches the shape creates a 100% unobstructed entrance to the carburetor and dramatically increases air speed. Take virtually any Dyno graph using any aftermarket production air cleaner available, and the power and torque gains will be obvious but much more significantly the power and torque becomes linear and smoother in comparison to the filter system its being compared to, hence a much smoother and crisper throttle from idle to peak rpm.” 4

Tim’s radiused aperture he calls a venturi is longer than any others I have seen in the industry. It lengthens the intake tract by one and a half inches, which is an awful lot and contributes greatly to the success of the Power PACC. The shape of the venturi is vital in that there cannot be any obstruction to undisturbed laminar airflow. The radiusing goal is to eliminate any restrictions to airflow whatsoever. In other words, the air directs to where it is supposed to go. Different

layers of air do not hinder other layers in any way. It is a cooperative effort with no turbulence inefficiencies.

Compare this venturi with the cast-in alloy radius incorporated into the S&S backing plate that lengthens the intake tract by one-half inch, which is a lot compared with many others.

Some Harley-Davidson backing plates have a flat metal or plastic backing plate with no incorporated venturi. Others integrate up to one-half inch venturis particularly on some Screamin' Eagle air cleaners.

The Doherty venturi is a work of art. It is a gorgeous, machined, and polished billet aluminum design that is all business. The venturi is so beautiful that it is a shame it sits inside the breather's machined and polished billet-backing

plate hidden from view. The Doherty venturi works as well as it looks.

Essentially Tim took one of the ultimate high performance air-feed intake assists, the velocity stack and put a high-flow air filter around it. The practicality of this all and the ease of installation makes this kit a winner.

Designed to handle up to 125 cubic inch displacement engines at 6500 rpm, the Doherty PACC air cleaners are compact for everything they achieve.

Doherty Power PACC Air Cleaner

Shown is the Doherty billet-backing plate with venturi installed and surrounded by his synthetic media high flow air filter. An air filter need not be an obstruction to producing

excellent performance gains. As Tim Doherty says, “The key to air filtration is controlling the air once it passes the filtering stage. I designed the Power PACC…to improve the airflow dynamics between the air filter element and the entrance to the carburetor or throttle body…Our venturi increases the overall intake tract length 1.5 inches (and) the shape creates a 100% unobstructed entrance to the carburetor and dramatically increases air speed…but much more significantly the power and torque becomes linear and smoother… hence a much smoother and crisper throttle from idle to peak rpm.” 4 Tim’s radiused aperture he calls a venturi is longer than any others I have seen in the industry contributing greatly to the success of the Power PACC. The shape of the venturi is vital in that there cannot be any obstruction to undisturbed laminar airflow. The radiusing must eliminate any restrictions to airflow whatsoever. The air directs where it is supposed to go. It is a cooperative effort with no turbulence.

Confusing Dyno Tests

I am going to compare three different Dyno tests for the Doherty Machine Power PACC air breathers performed at three different and reputable facilities in different parts of the USA. Competent technicians have performed the Dyno runs. Independent knowledgeable observers monitored the results for accuracy. Three different Harley-Davidson motorcycles were fitted with the Power PACC air breather for their individual tests.

Knowing the theory behind the Power PACC, it is no shock to me that all three independent Dyno tests on the different Harley's produced excellent to stellar horsepower and torque increases.

However, the power increases varied widely. So, why is this no surprise? This happens all the time without fudging

figures to make a product look better or worse than it actually is. It is very confusing for the consumer even when testing is above board. Like, who and what do you believe? Let us clear some of this up.

First, three different bikes were fitted with the Power PACC. The chassis's, so important in transferring horsepower and torque to the rear wheel differed in style, mileage incurred and efficiency. The bikes were of different years, with different modifications and most importantly different induction systems. One had a stock CV constant velocity carburetor, another had an S&S Super E butterfly type carburetor, and the third used a Delphi electronic fuel injection system.

Then three different Dynamometers tested the individual bikes for three different motorcycle magazines: American Iron, Easyriders, and Street Chopper. Therefore it is no surprise that resultant figures using the same Doherty air

cleaner design varied considerably but always on the positive side or should I say, very positive side.

Complete independent test results are on the Doherty Machine website

(www.dohertymachine.com) which reproduces the magazine reports.

Comparative HP and Torque Increases Gained with the Installation of the Doherty Machine Power PACC air cleaner 5

Three independent Dyno tests came up with torque increases from 2 ft-lbs, which is a good figure for the simple addition of an air cleaner to a whopping 11 ft-lbs, which is

stellar.

Horsepower gains varied from an extremely favorable 5 HP to an astral 10.4 HP.

My boss at American Iron Magazine, Chris Maida, the editor, monitored the first Dyno run on a 2000 Road King at H&L Performance in New England, which produced excellent results even though they were the lowest of the three tests. You cannot get anything by Chris who takes great pride in his integrity and getting out the truth whatever it may be. Chris related in American Iron Magazine, “And…how easy it is to install a Power PACC, the real star of the story was the amount of horsepower we got from this simple upgrade. Normally, we expect to gain a couple of ponies whenever we remove the stock H-D unit and bolt up almost any decent air cleaner. However, we were pleasantly surprised to see that we

actually gained about 6 horsepower, as well as almost 2 ft-lbs. of torque….The Power PACC also smoothed out the power curve for both the torque and horsepower, and delivered more power over the entire rpm range.” 6

Street Chopper Magazine recounted the testing of the 97 cubic inch S&S Evo at Lifestyle Cycles in Anaheim California. Street Chopper was very complementary to the Power PACC after the Dyno logged a 5-horsepower increase and an amazing 11 ft-lbs. of torque gain.

Easyriders Magazine wrote about the Dyno testing on a 2002 FXSTI with Delphi fuel injection and a Power PACC air cleaner at Grand Canyon Harley-Davidson in Flagstaff Arizona on May 17 2002. The wild increases generated by the Power PACC occurred in the lower rpm ranges where every hole-shot racer would want it. Easyriders relates that, “Initially, Doherty developed this unique system as a means of controlling air between the filter and carb…As one might expect, the

extension increases air speed velocity while reducing turbulence. Yeow! That's a whopping 10.4 horsepower increase and an additional 6.1 foot-pounds of honest torque… It appears that not only are Doherty's performance claims realistic, they may even be understated." 7

Tim Doherty proudly adds, "Easyriders magazine conducted an independent test for a technical article…The Power PACC was installed on a used stock 2002 Softail Standard with the center baffles removed that the dealer furnished. The bike base lined at 63.1 hp and 72.8 ft-lbs of torque. The dealer then installed the Power PACC and PowerVents with the following results 73.5 hp and 78.9 ft-lbs torque. The net gain was 10.4 hp and 6.1 ft-lbs torque. The Dyno operator and techs said the gains were not possible. They installed a Screamin' eagle air cleaner kit and gained 4.9 hp then reinstalled the Power PACC and achieved the same results within tenths. I would point out we had nothing to do

with this test whatsoever…Typically we see gains on a stock TC88 of 6 to 10 hp with 7-8 being the norm; however horsepower aside the cleaner crisper throttle response is well worth the price of admission by itself." 8

The Doherty Power PACC comes in two basic styles; the air cleaner kit itself and with gorgeously machined Doherty PowerVents that help control engine oil blowby.

They are available for

- Delphi EFI equipped bikes from 2001 to 2007 part number 01240 and 01240TK with power vents.

- Delphi EFI 2008 to present Twin Cams, part number 02148 that fit touring models equipped with 50mm fly-by-wire throttle bodies.

- Carbureted Twin Cams part number 01235 and 01235TK with power vents.

- Evolution, part number 01234 and 01234TK with power vents.

- Sportster, part number 01238 and 01238TK with power vents.

- There are also Power PACC's available for S&S E and G, CV and Mikuni carburetors that utilize the very popular S&S tear drop outer cover.

Doherty provides a billet machined backing plate with his radiused induction ring along with his air filter element with the Power Vents being optional.

## The Screamin’ Eagle Air Cleaner

I have often extolled the virtues of the Screamin’ Eagle air cleaner in the past. For a while, it was the only game in town and it was the first Screamin’ Eagle product that I liked. For years, it gave good solid gains in power…and it still does. It is just that so many other excellent products that literally give double the horsepower and torque gains have surpassed it.

### Screamin’ Eagle Air Filter Flow Rates

These are approximations since there are different filters used on various models.

## Do Atomizers Work?

This is a great time to discuss so-called atomizing add-on parts such as a fine mesh screen covering the mouth of the carburetor or fuel injection throttle body and rail.

The atomizer's purpose is to initiate turbulence by mixing air and gas before it arrives in the combustion chamber. I am skeptical whether these items work at all for a more efficient running modern engine although on the surface it seems so logical. Modern engines have combustion chamber design and shape to complement piston dome profile that very efficiently turbulate and atomize gas once delivered.

However, there is always the exception to the rule. In this case, I think the atomizers probably work well with hemispherical combustion chamber engines like Shovels, Knucks, Pans, and Ironhead Sportsters and Flathead chambers.

These engines have such inefficient mixture combustion chambers that only a low percentage of gas atomizes with the air in correct proportions to burn. We sacrifice faster and thus higher volume laminar airflow in the induction system on these hemi and flat head engines because what use is it if it does not mix well with the fuel in the combustion chamber. Any air and gas turbulence is welcome and needed on single-spark-plug hemi engines.

However, I will bet dollars to donuts that they do not work on a performanced engine with D-shaped or bathtubbed combustion chambers, featuring squish band(s) and mating pistons. In fact, they must surely impede airflow, which is what power is all about.

Remember that turbulent air moves much slower through the intake tract than laminar air. In this article, I will

repeatedly refer to laminar airflow into the combustion chamber. If wanting higher volumes of air reaching the combustion chamber we need less turbulence not more in this area of the induction system.

One of a performance air cleaner's main functions is to reduce turbulence in the air as it broaches the entryway of the induction throat. An atomizer will take the air that the air cleaner design has calmed down into laminar flow and then turbulate it, which will slow its travel and therefore volume.

Induction and port design delivers the air and the fuel to the combustion chamber. Here, Evo D-shaped combustion chambers, Twin Cam bathtubbed-shape ones utilizing piston dome mating-squish bands to force the air into a smaller chamber with great turbulence results in a much higher percentage of burn. In a modern engine with these features, we should leave the mixing to the combustion chamber design and volume delivery to the induction-port

H-D Delphi electronic fuel injection squirts the gas right at the intake valve backside. What use is an atomizer at the front entryway to the EFI fuel rail when the gas injectors deliver fuel directly to its final destination?

Not all will agree with this, but to me the logic seems inescapable. Atomization devices may work well for the wrong reasons on a carbureted hemi-engine but it is detrimental on its modern EFI bathtubbed counterpart.

This is also the reason dual plugging a hemi-head is a great idea whereas it won't have the same beneficial effects on a bathtubbed one. Hemi-heads need all the help they can get. For this reason, at Heavy Duty Cycles, we are inclined to smooth the intake port more on an EFI-bathtubbed engine we are porting and polishing whereas we leave the intake port on

a hemi-head rougher for turbulation and resultant atomization.

Even this will not have much effect as the layer of air/fuel closest to the port walls may have some mixture occurring because of wall roughness, there will be no affect on the inside layers at all. Those centralized streams of air will be moving in their own plane and at their own velocity.

S&S Air Cleaner

S&S Cycle, Inc. (www.sscycle.com), manufactures the most popular performance air cleaner of all time for Harley-Davidson applications. So popular, in fact that this air cleaner transcends its original function into aesthetic form also. Its styling alone is a reason for many to install the teardrop air cleaner on their bikes.

S&S Teardrop Air Cleaner Inner and Outer

The S&S Teardrop air cleaner is the most popular performance

one ever. The outer teardrop shape facing into the wind is aerodynamic thus reducing pressure drops and wind buffeting at speed. Shown is the inside of the outer cover with the inside of the inner mounting plate. The outside cover has a tapered protrusion on the inside that extends towards the throttle body inlet of the carburetor or fuel injection body. The air directional cone functions by smoothing turbulent air into faster and therefore, higher volume laminar flow. The inner back plate's radiused entranceway also maximizes airflow by creating a laminar effect. Turbulence in the induction system is the enemy. A seemingly simple air cleaner that increases performance and efficiency has much technology invested in it.

The radiused entryway to the carb/EFI throat blends smoothly in the track the air flows along to reduce turbulence.

The Teardrop air cleaner is available for stock H-D carburetors from 1967 to present including the Tillotson

(1967-1971), Bendix (1972-1975), Kehein butterfly carburetor (1976-1987), and the Kehein CV (1988-2006). The breathers also come for 2001 Softail and 2002, and up other Twin Cam models fitted with Delphi electronic fuel injection part number 17-0117; year 2006 and later Dyna models use part number 17-0444. Of course, they are available as originally designed for the legendary S&S proprietary S&S Super B, E, and G performance carburetors.

I asked the very knowledgeable Scott Sjovall, VP of Product Development at S&S Cycle, Inc. about the filter the Teardrop employs.

Scott told me, “The material we use has been carefully chosen to maximize the filtering ability while minimizing the affects of being wet with water or gasoline, deterioration with time and backfire flames.  All of this was balanced with

getting the lowest restriction, highest airflow possible. I firmly believe the filter element we are supplying with the E & G carburetor kits is the best on the market for peak flow while protecting for the real world." 9 Scott makes a point here that the reader should always employ when evaluating performance parts for their bike. Paraphrasing, he feels the S&S filter element is the best on the market for peak flow while it does its secondary but equally important job of protecting engine internals from dirt!

Now, this is important.

It is easy to make a faster street bike if there is no regard for engine protection and thus engine longevity. Protecting the engine is the compromise the reader must consider even though it may limit some power gain. How long does one want the engine to last before a rebuild? The performance work I do at Heavy Duty Cycles must pass one test, which is; can the bike ride from coast to coast without a breakdown? If

I feel this ability is jeopardized I will not perform the particular work unless apprising the customer fully and then getting the customers blessing to continue. For example, I will not install an unfiltered velocity stack on a customer bike until I know the customer understands all the ramifications.

The customer might be in love with the look, want maximum airflow for go-fast and all that with a box of chocolates but I want him or her to understand that the fuel intake will eventually gum up perhaps requiring on the road maintenance and the engine will wear out prematurely relative to one with an air filter. I have used velocity stacks myself but I understand the wallet-penalty and possible inconvenience that may occur.

Even in this book, the reader will see some figures that surely mean that one product is better than another is from a

performance perspective.

Two things the reader should ask themselves,

1. One, will the product be fully protective of the engine?

2. Secondly, is the part too productive for the size engine intended?

In the case of air filters, too much air for the size engine is just as detrimental as too little. The reader should also consider if the product offers equal protection to the stock component.

Of course, safety is always paramount in our minds also. For example, I would not be very responsible or caring for a customer's welfare by installing nitrous oxide on a mega cubic inch motor if the brakes, shocks, tires etc. were substandard

or in need of repair.

## S&S Fly-by-Wire Air Cleaners

At the time of writing, S&S does not offer the Teardrop for 2008-present 50mm fly-by-wire throttle bodies. The servomotor on the side of the throttle body necessitates a different backing plate. I am sure these will soon be available; see www.sscycle.com.

## S&S Hi-Flow Filters

I along with most have always liked the S&S air cleaner for both form (style), and functionality but S&S has gone a step further with the basic design to provide a much larger filter along with a cover that introduces even more air. I quote this venerable company's performance guide, "The new S&S high

flow filters is 1” wider than our standard filter and features a low restriction, pre-oiled filter element. The low restriction media, combined with the larger surface area provided by the one-inch wider filter, adds an airflow increase of up to 37% over our standard filter…The filter and filter adaptors extend the Teardrop housing out one-inch. The resulting band of exposed filter peeking out from under the Teardrop imparts a high performance street racer look.” 10 This kit, part number 17-0045, looks just plain tough. It will fit in existing E or G air breathers.

My suggestion also is to purchase the Patriot (#17-0385), or Desperado (#17-0004) air cleaner covers to provide a little ram airflow effect. If raining, I cover the openings with a little cover I made up. Inconvenient: perhaps, but what we will go through for some go-fast combined with good looks. The Patriot cleaner has an American flag motif cut out of the front of the cover while the Desperado has successive slots cut out for a nice styling touch providing a claimed one to two

horsepower increase.

The air cleaner housing has lots going on inside to channel air voluminously and efficiently, "…the uniquely designed backplate with air horn style radiused entryway and the dimpled cover with the air directional cone on the inside. This design helps maximize airflow by giving intake air an efficient, easy path to follow into the carburetor." 10 The radiused entryway to the carb/EFI throat blends the track smoothly that the air flows along to reduce turbulence. As I have discussed but it bears repetition, we want turbulence to mix air and gas once inside the combustion chamber but in moving as much air as possible in an efficient manner, turbulence becomes an impediment at this stage of induction.

The air directional cone, which is a tapered appendage extending towards the center of the carb/EFI entryway from

the inside center of the cleaner's outer cover basically takes turbulent air, smoothes it out and directs it into the entry way in a laminar mode. Furthermore Scott Sjoval adds, "The teardrop cover shape does a great job of moving through the air smoothly due to the outside aerodynamic teardrop shape, this helps minimize the pressure drops and buffeting effects when traveling at higher speeds. The shape also protects the intake from ingesting water during heavy rains while sitting still or while riding. This all goes together with the fact that it's a great looking air cleaner." 11

Comparative S&S Cycle Air Breather Airflow Rates

These tests were performed @ 28" H20 on a SuperFlow SF1020 on an EFI single throat throttle body and back plate mounted to flow bench by use of flow plate @ 28" H20 by Ann Christoffer, a Test Technician at S&S Cycle, Inc. on July

16, 2007. 12

*cfm is cubic feet per minute airflow

The filters used are not the traditional green foam S&S ones. Their newer version, features a pleated low-restriction synthetic media pre-oiled filter. The high flow filters are the same but one-inch wider allowing up to a 37 percent airflow increase over the standard pleated filter.

## Ram Flow Induction Systems

Ever since I can remember, Harley riders have had an innate desire for a big appendage extending from the manifold out and forward on the right side of the engine; it just plain

looks horny. Sometimes it is out and up as demonstrated by a downdraft Weber carburetor with velocity stacks extending its length up beside the gas tank. Once tuned, these exotic beasts could really cook.

Then, a decade ago, the craze was the dual-throat Delorto carburetor as this baby authoritatively bent forward alongside the engine with its extending air horns. They were easier to tune, for me anyway, the Delortos worked well. The Delortos then became very difficult to get and rumor had it that they could not pass emissions dictates.

Edelbrock (www.edelbrock.com) then followed suit with it's offering of dual Quicksilver carburetors that mimicked the look of the Delortos. The Quicksilver's look good and for a carburetor are more environmentally friendly. I always had a difficult time keeping the dual Quicksilver's tuned for any length of time.

Wimmer Induction

Companies like Wimmer Custom Cycle (www.wimmermachine.com), have developed a variety of good looking forward facing induction elbows that fit with velocity stacks from 2.5” to 3.5” long or with filtered stacks from 4.5” to 9.5” long.

Velocity stacks developed by Wimmer flow between 125 cfm and 290 cfm @10” H20 and the filters flow between 125 cfm and 290 cfm @10” H20.

This is a professional company as are the other mentioned sources that give flow bench test for cubic feet per minute flow capabilities of their products. Lee Wimmer does not live solely in the world of carburetion but also dwells in the

present and future. Lee expounds, “It has been my personal experience working with EFI that it is amazing what can now be done on a computer rather than screwdrivers and wrenches. It can be somewhat overwhelming but the same principles still apply. And to be very basic more fuel needs more air and you’ll get more power. Proper ratios need to be maintained, but the beauty of the new fuel management controllers is that you can easily make adjustments throughout the entire range very easily. If I were to give a plug for anything it would have to be Power Commander.” 13

Wimmer also provides the formula to calculate required flow rates for different cubic inch displacements at specified rpm’s at 10” H20.

rpm x cubic inch per cylinder x number of cylinders = cfm required

1925

For example, if we want to see the cubic feet per minute (cfm), ideal for a Twin Cam 88 at 2500 rpm we plug the figures into the formula to get our answer.

2500 x 48 x 2 = 124.67 cfm @ 10” H20

1925

If using a separate carburetor and air filter on each cylinder, the formula becomes:

rpm x cubic inch per cylinder = cfm required

1925

Each individual air breather will need to flow the minimum cfm given by this formula.

A Harley filter flows approximately 130 cfm at 10” H2O. We have some room to spare. As I have discussed, a flow bench measuring cubic feet of air through an air breather uses a manometer, which is a tube that fills with water to determine the test pressure used. The pressure of the air will displace ten inches water into the manometer.

However, if we use the formula to find out cfm for 4000 rpm we will need a cfm rated filter for 200 cfm. We now have a restrictive air filter. The Screamin’ Eagle filter rates at approximately 160 cfm at 10” water, which is much better but still restrictive in this example. Wimmer list filters that flow up to about 230 cfm at 10” water.

Wimmer also supplies performance induction elbows,

stacks and filters for

Magneti-Marelli (MM), electronic fuel injection used from 1995 to 2002 on various Big Twins and Twin Cams. There is not that many companies that pay attention to these bikes because the shape of the MM EFI entryway is not round but is a rounded rectangular shape.

In talking to Lee at Wimmer, it is obvious that he takes great pride in the quality and number of options available. As Lee says, “No one offers more fitment than us. We have designed intakes for almost every application out there assuring you get a professional fit and performance. We take great pride in manufacturing EVERYTHING in the great USA. We are committed to making constantly new styles for both looks and performance.” 14

With Wimmer products, form, once again meets function in a perfect marriage of good looks and performance gains.

Wimmer Spike Stack on a 900 Runner

The Wimmer Spike Stack machined from 6061-billet

aluminum chromed to a brilliant finish is beautiful as well as functional. The internal cotton filters come in four colors; black (shown), blue, red, and purple to match and complement any paint job. The Spike Stack shown on a forward runner will also fit on their runner or their dual and dual runners. Top breathers are available with all Wimmer Twin Cam air systems. The gorgeous pictured radius-groove (smooth and Maltese Cross designs are also available), billet units have a reusable filter element that allows for maximum breathing while trapping and holding excess blowby. Wimmer has an abundance of performance filters that marry to basic build intake kits that fulfill everyone's version of "cool". For the racing drag bike set, there is a velocity stack for every need and taste.

Wood High Flow Air Filters

Bobby Wood of Wood Performance Carburetors (www.woodcarbs.com), and more recently of his Knight

Prowler camshafts also has his own line of King High Flow Air Filters.

Developed to Bobby's specifications by K&N Engineering (www.knfilters.com); K&N manufactures these filters exclusively for Wood Performance. These maximum flow filters will handle between fifty through 140 HP. The figures given are flow tested at 1.5" H2O. Designed for fifty to 78 HP is the King Filter KF250, which flows 220 cfm at 1.5" H2O; the KF275 is for seventy-eight to 90 HP and flows 250 cfm; the KF 300 is for ninety plus horsepower and flows 282 cfm. The KF325 flows 340 cfm at 1.5" H2O is for the big cubic inchers. Compare these figures to a Screamin' Eagle filter that flows 135 cfm at 1.5" H2O.

Wood Performance Carburetor Air Filter cfm flow rates @ 1.5" H2O

Bobby Wood also supplies a big eight-inch chrome performance cover for his filters with big slots cut into the sides for air ease of access. He also has chrome billet air filter covers that leave the filter pleats exposed to the air for maximum flow capabilities. As Bobby says on his website, “When you need more HP and Torque…you must add more fuel and air.” 15

S&S Single Bore Tuned Induction Systems

The S&S tuned induction system is available for Magneti-Marelli electronic fuel injection used on 1995 to 2001 Big Twins and Twin Cams and for the much superior Delphi EFI, used from 2001 to present. S&S supplies both single bore and dual bore induction systems for various applications where one or the other will fit Delphi, Magneti-Marelli and the S&S

proprietary VFI fuel injection systems as well as Super E and G and the stock H-D CV carburetors This bolt on kit looks trick, very trick, and gives improved power and torque during the whole rpm range.

What Does Donny Use on his Bike?

I have the S&S single bore tuned dual filtered induction system fitted on the larger 50mm EFI throttle body on my current ride, a 2007 FLHXI. It is also bored out to 103 cubic inches powered by the Wood Knight Prowler .650" lift TW408G gear driven cams that in turn are powered by Bob Wood's Advanced Belt Drive System with Screamin' Eagle heads, HTCC forged 10.5:1 compression pistons, roller rocker arms with all these modifications tuned with a Race Tuner and she honks. I later replaced the Race Tuner with a Power Commander for ease of tuning. I use the S&S stainless steel

Bagger mufflers.

Two Jagg 10-core turbulated oil coolers reduce engine temperatures (www.jagg.com), on my engine. This addition will add engine longevity and goes a long way to eliminate any engine detonation by lowering cylinder head temperatures.

This parts combo is not the secret formula to the fastest bike; it is just my current one of hundreds of formulae used over the years. However, it is a good one and I will be happy with it until the next idea pops into my brain. Before the reader installs any of these parts, I would like to clarify a couple of things. I am very happy with Wood products, as I am with some other cam manufacturers including Andrews (www.andrewsproducts.com), Crane (www.cranecams.com), and S&S Cycle, Inc.

However, I like the torque provided with the very high lifts relative to duration and overlap of Bobby Wood's camshafts. And ya just gotta love the look of his Knight Prowler Advanced Drive Belt System. The carbon fiber belt running drive pulleys for the gear-driven camshafts is visible behind a clear cover surrounded by lustrous chrome.

Harley-Davidson is looking for ways to reduce noise; well, here is one.

Sometimes with other brand cams that have longer durations relative to lift I will use increased ratio rocker arms to effectively increase lift. Stock rocker arm ratio for the Twin Cam is 1.625:1. Multiplying effective cam lift in the timing case by 1.625 will give real cam lift figures in the heads that open the valves. Ergo, if using, say a 1.675:1 rocker arm ratio, the effective cam lift now multiplies by 1.675 for a higher real cam lift at the valve.

If one knows which Screamin' Eagle heads and piston combinations to choose (www.harley-davidson.com), they make a good choice but again so do other head-piston combinations depending on application. The trick is to know the quality manufacturers from the others and then to choose the right combination of performance parts for style of bike and type of riding desired with reliability concerns overriding the choices I personally make…it is all about the power band.

Jagg is the only oil cooler for me. In my opinion, there are better-looking coolers than Jagg but none work better. While the H-D coolers, everything being equal, cool as well as a Jagg, they are more restrictive. I do not like more restrictive. I cannot double up on more restrictive oil coolers, as the restrictiveness will also increase, so beware. If you are like me and want ultimate cooling to stop the great engine destroyer,

detonation, the restrictiveness of running two H-D coolers may not be a good thing. These restrictions are not a concern with the Jagg coolers.

I use a few favorite ignitions. I will not touch many more with a 10-foot pole.

I am currently trying the race tuner and have not made up my mind yet on how much I like it but so far so good… although the Power Commander seems better to me right now. In my opinion, Power Commander is user friendlier. However, that may change as technology marches on.

I also use and like the self-tuning Terminal Velocity (www.terrycomp.com), Power Commander (www.powercommander.com), and S&S ignitions to name a few. There are new ignitions coming on the market all the time so this list may become time dated.

I did not want to split the engine casings on this particular bike since I am fooling around with the new gear ratios etc. found on the 2007 Twin Cams. If I wanted more power then stroking and bigger bore would be the next step. Why fool with a 110” engine when the same amount of work and money will get you a very reliable 124” S&S mill in your stock H-D engine

Enough of me, what do you think of the very cool S&S single bore induction system I am using?

S&S Dual Ram Flow on Donny's Current Engine

This is not the magic blueprint to power although it is I do up a different engine for myself every year checking out the

flavors of the day. My current ride is a 2007 FLHXI bored out to 103 cubic inches utilizing Screamin' Eagle heads with HTCC forged 10.5:1 compression pistons and tuned with a Race Tuner. I am using the S&S single bore tuned dual filtered induction system fitted on a larger 50mm EFI Delphi throttle body. The Wood Knight Prowler .650" lift TW408G gear driven cams driven by the Master of Torque, Bob Wood's Advanced Belt Drive System, powers all this. Two Jagg 10-core turbulated oil coolers reduce engine temperatures out of detonation range.

The Forward Tuned Induction System

The stylish S&S forward swept design does not interfere with a crash guard if using one. More importantly, from a safety and comfort perspective, the rear brake is easy to use, as the dual independent tuned-intake tracts do not impede the rider's leg whatsoever. Many of the trick systems that stick out and forward forces the riders leg out and around the system making braking awkward at best and dangerous in a crisis situation but not so with the S&S system.

Scott Sjovall explains further, "The major three benefits of the single bore induction system are;

1) It moves the air filters forward for more leg clearance

2) It utilizes two filters that have more surface area than the standard element in the tear drop cover

3) The long runners give a larger plenum volume of filtered air, ready for the next intake stroke.” 16

Standard air cleaners and air intake depend largely on the vacuum or negative pressure created when the pistons descend in their respective cylinder bores to draw in the air for the new fuel charge. This forward tuned induction system’s larger filter areas allow more air through; there is a larger air supply waiting for the next intake stroke, and the air compacts from pressure behind by more air supercharged by the ram airflow effect. The air will pressure in at a higher rate, which is just what we need for more horsepower and torque. The S&S Cycle 2007 Performance Guide states that, “The Tuned Induction systems are for riders who are seeking

the greatest possible performance gains from their fuel injected engines. These scientifically designed induction systems feature two tuned intake tracts...feature carefully chosen lengths, diameters and tapers. By using air column inertia and carefully timed gas dynamic pressure waves...ram additional air into the cylinders for increased power. The most pronounced effect of this natural “supercharging” effect is increased midrange torque...right where the street rider needs it most. The forward swept design of the intake runners further improves power by providing the engine with cooler, denser air.” 17

Cooler air is denser providing compacted oxygen molecules. Hot air expands and is less dense with less oxygen. There are big advantages with this system in that there is more oxygen rammed in higher volumes.

Oh! Did I mention the good looks?

# Flow Bench Manometer Test Pressure Conversions

The standard automotive norm for flow testing air filters on a flow bench is to use a manometer test pressure of 1.5”. The industry leader K&N uses this test pressure as does Bobby Wood of NASCAR fame at Wood Performance Carbs.

Head flow testing is usually the domain for higher manometer test pressures.

Once we know to evaluate flow test results for different air filters, heads, carburetors, or exhaust pipes at the same manometer test pressure in inches of water we can start to have potentially meaningful comparisons. On the surface, one might think this is all that is required to determine the top guns.

When reading ads that espouse a certain product, ostensibly proving their assertions with cfm flow rates it is at least necessary to know the manometer water-test pressure. An ad for another air breather may seem much better until reading the test pressure. If the manometer water-test pressures are different, we need a formula to equalize the figures to begin making a fair comparison.

The formula is:

The square root of (wanted test pressure ÷ known test pressure) x cfm

As an example, if we have a known test pressure of 300 cfm @ 18” H20 what would the airflow cfm would be at 12” H20?

Square root of (12 ÷ 18)

Square root of .667 = .817

300 cfm @ 18” H20 x .817 = 245 cfm @ 12” H20

Generally, as a rule, a lower manometer H20 test pressure cfm airflow will convert and be equal to progressively higher airflow cfm rates with higher test pressures. Conversely, higher manometer H20 test pressure cfm airflow will convert and be equal to progressively lower airflow cfm rates with lower test pressures. I have developed a conversion chart by the above formula so the reader can more easily make comparative calculation.

Flow Bench Test Pressure Conversion Chart

If I want to convert a known test pressure of say, 250 cfm @ 12” H20 to the equal cfm rating at 18” H20 I would multiply 250cfm by the chart factor of 1.22 to obtain 305 cfm @ 18” H20. Therefore, 250 cfm @ 12” H20 is the same as 305 cfm @ 18” H20 all other factors being

A Flow Bench is a test tool that measures cubic-feet-per-minute (cfm), airflow through an opening such as an air filter, air breather, head, carburetor venturi, exhaust pipe, or any other aperture. Air is required for power. Gas cannot burn unless mixed proportionately with air. Air (exhaust gases), must also exit an engine to allow fresh air in. Therefore, airflow is a key component of performance work as well as engine efficiency.

Often performance and efficiency are symbiotic and even

synonymous because making an engine more operate more efficiently will make it faster as a by-product.

An advertisement giving an impressive airflow cfm rating to promote a particular product must also give the test pressure used to obtain the cfm figure given. Otherwise, it is meaningless especially when comparing the figure with that for a competing product. Furthermore, this conversion table can be dangerous and very misleading when converting test pressures in an attempt to obtain accurate comparative figures.

Some reasons are as simple as,

- Is just the air filter or is the whole air breather assembly tested.

- Has the same attachment fixture been used on the

different testing flow benches?

- Have the different flow benches been calibrated properly?

- Have the same test procedures been used?

- Has barometric pressure been factored in?

Also remember there are three main factors determining airflow into the induction tract,

1. The aerodynamic design of the breather.

2. The flow rate of the filter itself.

3. The design, if any, of the radiused aperture that will convert turbulent air into high volume laminar flow.

I have given published cfm airflow figures for various products in this article. These are all very reputable businesses; there are many more highly professional companies on top of their game not mentioned because of space constrictions.

It is not a good idea to use this chart or the conversion formula to see which competing product from different companies is best. However, it is probably accurate to compare products within the same company since one would think they have used the same testing parameters. I have not used the formulae or the conversion chart, in an attempt to find the supposedly best product. The reason is that I cannot and nor can you with the minimal known information. I simply do not know all the test variables. I can assure you, all products listed in this article are excellent as are some others

not listed.

In addition, one product may work better in one engine application and not as well on another. One or two superstar components do not make a fast engine. Coordinated and complementary teams of parts that accentuate the abilities of the others and vice versa produce the fastest Harley-Davidson.

## Carburetion

Carburetors are all about pressure differences whether caused by the weight of air (atmospheric pressure) or by vacuum. Vacuum may result from a venturi design basing on Bernoulli's Principle or by rapidly descending pistons or by higher rpm scavenging also known as the physics Principle of Momentum. Without these differentials in pressure, carburetors would not function at all.

Furthermore, to state the obvious; gasoline cannot burn by itself. Gasoline needs to combine with air in order to burn.

Pressure differences direct airflow as well as vaporized fuel via principles of physics that cannot be changed or violated.

Internal engine design changes rapidly in tandem with ever increasing governmental emissions dictates that attempt to achieve complete combustion. More air relative gas very effectively decreases harmful emissions.

The Butterfly Throttle Plate

Most motorcycles use one of four types of carburetors.

1. Butterfly throttle plate (Bendix, S&S).

2. Slide control with throttle cable activation (older Mikunis).

3. Vacuum operated slide control with throttle operating the butterfly plate (CV Kehein).

4. Slide control with throttle linkage (Mikuni Smoothbore)

The H-D butterfly carburetors include the Schebler, the famed Linkert, the rare DC Linkert, the much-maligned Tillotson, Bendix, and the Kehein not to be confused with its successor the CV Kehein. Butterfly type carburetors like those listed above powered Harley-Davidson's from the early nineteen hundreds until 1988. The aftermarket supplies many types also like the S&S Super B, E and G butterfly type carburetors.

These carburetors use a butterfly, vernacular for a flat circular disc that pivots in the carburetor throat to control airflow through the venturi. A throttle cable connects directly to the butterfly shaft and lever on the outside of the carburetor. When the butterfly closes, airflow severely limits, which stops any venturi action. The butterfly is not set to close 100% because a little air is required for idling. As the twisted throttle opens the butterfly, air and fuel correspondingly increase in flow to create more engine power.

## The Venturi

Publications always refer to the venturi. However, no one ever seems to explain what a venturi is or the physics principle that it relies on. The venturi is part of the carburetor throat. The throat is the opening through the centre of a carburetor.

Bernoulli's Principle comes into play here. Mechanics may call it the Venturi Principle as a descriptive way to get away from the guy's name that first recognized this universal physics principle.

A venturi is a restriction or more correctly, a narrowing formed in the carburetor throat that causes an increase in the velocity of air. Think of a wide lazy river as it narrows. The water must speed up in order to pass the same volume through the narrowing in the same amount of time. If the narrowing becomes a wider avenue, the water will slow down as the same volume can pass in the same amount of time. If I describe this in another way, the increase in air/fuel velocity through a venturi also causes a vacuum increase (decrease in pressure), as the air passes through.

This pressure difference draws fuel through jetting from the connecting float reservoir and into the air stream. If looking at slow moving air (low vacuum), on one side of the carburetor throat, it speed up as it must get through the narrower venturi in the same period of time thus creating high vacuum or suction, which draws in the gas.

The gas begins to atomize as it draws into the fast moving airflow. After passing through the venturi, the airflow slows down as it enters the manifold and into the intake port to draw into the combustion chamber via another vacuum created by descending pistons.

Valve overlap creates a different origin vacuum at higher rpms that draws in the fuel mixture into the combustion chamber via the physics principle of Momentum also known by the mechanical term scavenging.

The second type of carburetor, which is not in use on a Harley except in the aftermarket, is a slide control. The throttle cable activates the slide. The older Mikuni's are a good example of this.

The third type is a vacuum controlled carburetor, which uses both a slide and a butterfly to control airflow through the venturi. The throttle cable does not attach to the slide but to the butterfly. Vacuum acting on a piston and diaphragm operates the slide on these carburetors like the CV Kehein. The vacuum piston controlling the size of the venturi opening is the difference with the CV carburetor. This is a precision mechanism that controls air and gas flow via vacuum.

The sole butterfly semi-rotating plate methodology located inside the venturi on H-D predecessor carburetors is wasteful, inefficient, and disruptive to airflow.

CV Kehein Carburetor

The CV is a wonderful carburetor but not as efficient as its successor, electronic fuel injection (EFI). However, it is one of the few carburetors that attempt a response to altitude changes. It is partially successful in that the pressure difference in the upper and lower chambers reduces as elevation increases. Therefore, less fuel delivers because of the pressure reduction.

As I will mention many times, as altitude increases, atmospheric pressure decreases.

Decreased atmospheric pressure is the fancy way of saying there is less air available. Conversely, the pressure difference in the two chambers, upper and lower, will increase as elevation decreases towards sea level. More fuel will deliver

because of the pressure increase.

It is time to describe the CV carburetor. It is a constant velocity, gravity fed carburetor with a float operated inlet valve, a variable venturi via slide action with a throttle stop screw for idle speed adjustment. There is a fuel enrichment system for starting. Idle and transfer ports provide a balanced fuel mixture during the transition period from idle to mid-range. A vacuum piston controls the venturi opening and the CV has an accelerator pump.

The accelerator pump system uses sudden throttle openings (rapid acceleration), in order to rapidly squirt fuel into the carburetor venturi to provide the extra fuel necessary for smooth acceleration. Aggressive throttle twisting activates the accelerator pump. However only the first third of complete throttle travel pushes the accelerator pump rod

down, which flexes a diaphragm.

Earlier model H-D carburetors outfitted with accelerator pumps use a rawhide or rubber cup as opposed to a diaphragm to force or squirt the gas into the venturi in a steady airborne stream. The check valve stops any back flow of gas during this downward accelerator-pump-rod-stroke. A spring returns the diaphragm back into its original position. Gas then flows from the float bowl reservoir underneath the diaphragm via pressure differentials in order to replenish for the next handful of throttle twisting.

Carburetion gas delivery is through another physics principle, gravity from an overhead-located gas tank.

The choke butterfly plate that restricts air into the venturi in order to provide a richer mixture (more gas relative to air)

in all previous H-D carburetors is crude when comparing to the CV enrichment system. Like the operation of most butterfly choke plates, the CV enrichment circuit controls by a knob and linking cable that locates under the gas tank on the left hand side of a Harley while sitting on it.

The more sophisticated EFI does not need either enrichment or choking systems as the computer microprocessor dictates pressurized-fuel needs via information provided by sensors.

Early usage of the CV did not involve a comprehensive tie-in sensor feeding microprocessor controlling engine management system. Today's carbureted Harley's until their last year in 2006 has engine management that is very sophisticated but still cannot achieve the levels of fuel economy, emission control and even performance that later Delphi EFI broaches.

The radiused entryway to the carb/EFI throat blends smoothly in the track the air flows along to reduce turbulence.

CV assisted cold starting evolves from the use of traditional and crude butterfly choke plates to a more sophisticated enrichment circuit to the current ultimate sensor fed computer. The enrichment knob can pull out incrementally to a full position depending on temperature and altitude. Manuals instruct not turn the throttle at all when utilizing a carbureted enrichment system. Nice try! Harley riders on carbureted bikes will never seem to learn not to touch the throttle.

Surveys report within a margin of error of plus or minus one that nine out of ten Harley riders surveyed believe throttle blipping is a necessary ritual act of domination. Those

riders that have EFI equipped Harley's quickly learn not to blip the throttle as they patiently watch for the fuel pump EFI indicator light to go out that locates on or near the speedometer face.

Manuals instruct all kinds of rules for how long to leave a carburetor enrichment device on. They have instructions for hot engines, for cold engines, instructions combining with distance rode, and/or time to leave the device on. My suggestion is to use these instructions as a starting point if you are a novice H-D rider. However, these seemingly inanimate machines all have their own peculiar personality traits. Starting procedures will vary from bike to bike. Experiment and the bike will not be afraid to voice its displeasure until it purrs in a potato-potatoey sort of way. If the enrichment knob is in the wrong position the bike will immediately react by idling poorly either because of not enough gas or too much gas.

The same goes with acceleration and cruising. Why is this so complicated when the system is so sophisticated? Lean emission dictates are fooling with the laws of physics so they must compensate with more gas (enrichment) to combat too much air. These engines do not want to run initially because the air/fuel mix is way off. Once the engine warms to operating temperature, it is easier to ignite these super lean (more air relative to gas), mixtures.

On my own bikes, I pull the enrichment device full on, start her up, and begin to push the knob slowly back in. If the bike stutters, I pull it back out. If it is bogging down, I keep slowly pushing the knob in turning the Enrichener off. It is no big deal; after awhile, it becomes second nature.

The enrichment (starter valve) device meters fuel upward through a valve chamber.

The starter valve opens a passage into the carburetor venturi on the vacuum side. Air also directs to the valve chamber to mix with the gas. The descending pistons on their downward intake stroke cause the vacuum (low pressure). Gas from the float bowl chamber will draw up into the carburetor venturi by the pressure differential. There is lower pressure in the venturi because of the descending pistons, which causes the now higher pressure in the float bowl chamber to force the gas to ascend into the lower pressure area.

## Ignitions Coordinating with Carburetion

The electronic ignition systems introduced in circa 1978 forms the basis of the present day engine-management-systems. Certainly, the original electronic ignitions were crude relative to what is standard fare today and had interesting but extremely frustrating teething problems.

Point's ignitions have moving parts while electronic ones are solid state with no moving parts. The electronic successors are virtually non-adjustable without rebuild possibilities. The tinkering Harley riders of the day hated them for these reasons.

The V-Fire II corrected some of these problems. However, electronic ignitions did not start to have much efficiency over the previous points ignition systems until the V-Fire III's introduction circa 1983. This electronic control module functionally co-ordinates engine management.

The V-Fire III utilizes a Vacuum Operated Electric Switch (VOES). So, what does all this have to do with a carburetor? The connection revolves around the VOES switch, which mounts into the intake manifold. High or low vacuum

conditions in the carburetor venturi are an indicator of engine load. Engine load is how hard the motor is working. Rapid acceleration and riding up hills would be examples of heavy or increasing load while cruising leisurely on the highway is a no-load situation.

Because vacuum parallels engine load, its use can match ignition timing to the engine's needs. The VOES also known as a vacuum advance unit, advances or retards ignition timing via the V-Fire III ECU (ECM) in order to increase both power and fuel economy. For example, under heavy loads, vacuum in the intake manifold will drop and correspondingly retard ignition timing. A vacuum hose connects the VOES switch to the carburetor or manifold internal opening or venturi through which gas and airflow. Electrical wiring connects the VOES to the ECU microprocessor.

The V-Fire III programs with two spark curves to delineate varying engine load. The ignition curve associated with high carburetor venturi vacuum sets the maximum advance to ignition timing. The low venturi vacuum curve retards ignition timing. Retard spark ignition assists in reducing detonation (spark knock), while providing performance under high-load conditions like acceleration.

## Intelligent Spark Technology

S&S Cycle, Inc., (www.sscycle.com), IST Intelligent Spark Technology ignition system is proprietary technology developed by this venerable aftermarket Harley-Davidson performance company that is now branching into its own 2010 EPA compliant engine, the X-Wedge and performance parts for Victory Motorcycles.

This carburetor integrated ignition system is automatic. It

gets better, as the IST adjusts ignition timing via an anti-knock sensor. Ignition timing automatically advances for maximum power unless the IST senses detonation, in which case the ECU retards timing until it disappears. The IST monitors the crank position sensor (CKP), the cam position sensor (CMP), if the engine has one, the manifold absolute pressure sensor (MAP), cylinder head temperature sensor (ETS), as well as the exclusive S&S anti-knock sensor.

The IST has a memory that recalls situations where detonation occurs and adjusts timing appropriately. Additionally, the memory learns the particular engine's ignition requirements it is timing and writes a custom ignition map for it. Even so, the IST will override its own map if a parameter changes such as filling the gas tank with a lower octane gasoline. Furthermore, if further engine modifications occur such as hotter camming, stroking, big bore, exhaust, carburetion, cylinder head modification, or whatever, the IST

will reconfigure ignition requirements automatically.

The standard 6250 rpm rev limiter is adjustable up to 7200 rpm maximum.

IST is single-fire. An automatic dwell adjustment will optimize current for any coils 0.5 ohms to 3ohms. It protects against reverse polarity and short circuits.

Who would ever think carburetion could integrate with ignition timing so seamlessly? IST is accordingly so good at what it does that S&S extends warranty on all their carbureted big-inch engines when incorporating this ignition system.

## Tuning the CV Carburetor

Carburetion on a Harley-Davidson is all about compromise. It is rarely perfect because of barometric influences and the

overlapping of various fuel circuits where each has a specific primary function.

## The Enrichener Circuit

The Enrichener circuit, which may also be a butterfly choke plate obstructing the carburetor mouth, richens the AFR on startup to aid in cold starting and engine warm-up.

## The Accelerator Pump

The accelerator pump squirts fuel into the carburetor throat when first twisting the throttle to also assist in starting and the transition area just off idle.

## The Idle and Low Speed Circuit

The throttle closes at idle. Fuel metering through the slow-speed jet and a mixture screw controls idle speed when the throttle plate is barely open. At low speed when the throttle opens the plate a little more additional fuel directs into the carburetor throat venturi. The lowered position of the vacuum piston reduces the venturi opening. The idle and low speed circuit controls idle and assists the transition into mid-range. This circuit controls 100% of idle and about 25% of right off idle low speed. However, the primary function of this circuit is the idle.

The Mid-Range Circuit

The mid-range circuit-slide position rises from the lower low speed location. This movement occurs as the throttle plate opens creating a low vacuum situation that moves the piston upwards. The throttle is partially open and controls via a

tapered needle attaching to the piston that protrudes down inside a tube. As the piston raises so does the needle out of its mating seat: the needle jet, allowing fuel to rise into the air stream to atomize.

The mid-range circuit controls steady throttle opening and lighter acceleration.

The High Speed Circuit

The high-speed circuit activates by opening the throttle plate further creating a pressure differential, which raises the piston slide fully upwards opening up the venturi to its full size. Vacuum controls the slide that lifts the needle all the way up with wide-open throttle during hard acceleration. The high-speed circuit controls the amount of gas passing by via a main jet that the tapered needle slides up-and-down. The

needle jet opens to its maximum allowing in enough fuel to meet engine demands.

In this situation, of wide-open throttle during hard acceleration the main jet is the primary controller of fuel delivery. The needle will also control fuel for everything in-between idle and full tilt boogie. It will start to fuel right off idle with light throttle. As the throttle opens, air velocity increases under the slide and fuel will draw up from around the needle, even before the slide begins to lift.

## Basic Carburetor Tuning

Do not confuse other carburetor or mechanical problems with a jetting situation. For example, a tuner can jet forever but this will not resolve a severe manifold air leak. The tuner may be able to adjust jetting to resolve a minor air leak by compensating more gas for the extra air intake but this will

only create problems in other areas that will become very frustrating.

First, tune each circuit for its primary function to create a base line.

1. Adjust the idle jet and fuel mixture screw for the best idle.

2. Set the main jet for best wide-open throttle.

3. Regulate the needle for steady cruise with light throttle.

It can take a Harley twenty miles of riding in order to reach operating temperature for fine-tuning purposes. In addition, temperature and air density can vary greatly from

morning to afternoon sun to dusk

If experiencing, jetting related ride-ability problems, attempt to replicate the problem in all gears and observe during temperature changes day-to-day or morning to afternoon. This will provide clues as to which circuit needs tuning whether it is idle or light throttle, which will require slow speed jet, mixture screw adjustment; steady throttle and/or light acceleration will require a needle correction, and finally wide open throttle/hard acceleration will need a main jet adjustment.

Decide if the carburetor is running with too rich an AFR ratio or a too lean AFR.

- If the problem is aggravating when hot, this is a too-rich symptom.

- If the problem is irritating when cold, this is a too-lean symptom.

- If the situation worsens in higher gears, this is a too-rich symptom.

- If the situation worsens in lower gears, this is a too-lean symptom.

- Make only one adjustment at a time to isolate the problem!

- After making an adjustment, attempt to duplicate the problem again.

There are tricks that may make the tuning process easier

and faster. For example, if suspecting a main jet mal-adjustment it will probably be easier to test for a rich or lean condition first. In this case, remove the air filter if you think the jetting is too rich. More air entering the carburetor will lean out the rich condition.

Duct taping up to a half of the filter surface area of the air filter surface if suspecting a lean condition. Less air entering the carburetor will richen the lean condition. Sometimes just removing the air filter and holding your hand over the carburetor throat will steer the tuner in the right direction.

Once knowing the direction of jetting change or adjustment, the severity of the problem will indicate a starting position for the adjustment. These simple tests may complicate another circuit temporarily. The tuner should ignore these symptoms, as they are not permanent.

A common complaint is an off-idle pop or backfiring out through the carburetor at slow engine speeds. I immediately advise the owner to assure that there is not a manifold air leak, as this will also cause this annoying and dangerous problem. If not a manifold leak, this is usually an excessively lean idle mixture. This easily corrects by up jetting.

I like to do the jetting in two stages; first, perform the idle and low-speed jetting and then proceed to the high speed or main jetting.

I will be describing spark plug readings, so it is imperative that with this methodology the ignition timing is correct. In addition, the ignition system must be functional up to manual specifications.

A weak yellow or non-existent spark will drastically alter spark plug readings. Sparking across the plug electrode should be crisp and either white or blue in color.

I am going to use the CV Kehein for my example. Adjustments for the S&S Super E, the HSR Mikuni Smoothbore and other carburetors in popular use will be different. Consult the manual for the particular carburetor in service. I will be discussing these carburetors further on in this chapter in the section titled, Tuning the S&S Super E and G Carburetor and in section, Tuning the Mikuni Smoothbore Carburetor.

First, I need to get the idle and low speed jetting correct and then I work on the high speed or main jetting. Before starting, have a collection of fresh unused plugs gapped and ready to use. I will need at least three new sets for this test. I

favor Champion plugs for both plug readings and for everyday use in Harley-Davidson. Mechanics have differing luck with different brand spark plugs. One thing is for sure; Champion is less expensive than H-D spark plugs. I also think Champion has private-branded spark plugs for the Factory over the years. EVO Big Twin use Champion RN12YC. EVO XL and TC88/96 use the Champion RA8HC. Set plug gap between .038” and .043”. There are premium spark plugs of almost every brand that will serve a performanced Harley admirably but that is a subject that will be covered in Volume VI: How it all

On the CV carburetor, adjust the idle mixture while idling. Closing the idle mixture screw slowly will cause the idle to become rough. Slowly turn the screw out until the engine again idles smoothly. Add approximately ⅛ to ¼ turn more. If closing the screw makes no difference in idle speed or smoothness, replace the pilot jet with one size under.

Most Big Twins use pilot jet #42 as the stock one until the 2000 model year. Year 2000 models use #45 for stock application. Consult the workshop manual for year and model. This presents an irritation because H-D applies their part numbers to these and other jets. I usually cross-reference them on the Internet or with an aftermarket parts manual. In some cases, depending on the bike set-up and higher altitude, #45 may be too large. An aftermarket #44 pilot jet works very well under most conditions. If finding it necessary to turn the screw out more than three turns, increase the pilot jet size to the next larger size.

After setting the idle speed and idle mixture correctly, change to a fresh set of spark plugs. After fresh spark plug installation, ride the bike at part throttle, gently accelerating but keeping below ½-throttle. The goal for this stage of tuning is to stay within the low speed circuit as much as possible. After a mile or more, pull in the clutch, press the kill button

and coast to a stop on the side of the road. Using the glove and plug wrench from your back pocket, remove the hot sparkplugs. Examine the insulator from the tip down as far as is visible.

- The correct color is white without blistering, off white or off-white with a very slight tan tinge.

- Blistering indicates a too-lean AFR condition and is dangerous.

- A tan, brown, chocolate brown or black fluffy color indicates a progressively rich to very rich AFR condition that creates excess fuel consumption and a potential power loss.

- Flat black fluffy residue is always a gas-fouling situation.

- Shiny scaly black residue is certainly oil blowby fouling.

Either a gas or oil-fouling situation can easily prevent the bike from starting or running.

It is normal for the rear cylinder spark plug to read slightly richer than the front. The reason for the difference is inherent in the design of the cylinders and the intake manifold design. The correct procedure is to change jetting up or down basing on the front cylinder spark plug reading. Up jetting corrects a lean condition, whereas down jetting in size corrects a rich AFR condition. Before changing pilot jets try to richen or lean the mixture by turning the mixture screw in to lean a rich AFR condition or turn it out to richen a lean AFR scenario. Once the plug reading is acceptable, proceed to the main jetting.

Main jetting by spark plug reading requires a highway since safe acceleration at wide-open throttle through the gears up to and including fourth gear is necessary. Fifth gear is theoretically better but wide-open throttle will take this diagnostic testing into illegal and sometimes dangerous speeds. The procedure is simple; change to a fresh set of plugs, start the bike and immediately accelerate at wide-open throttle through the gears including fourth gear. Idling or using part throttle will bastardize the plug readings and possibly negate the test. After accelerating through fourth gear, pull in the clutch; push the kill switch and coast to a stop. Remove the spark plugs. If the rider forgot to put on a glove for the first test(s), forgetfulness surely will not be a problem this time around. Examine as before and follow the same procedure with the main jet.

- If the plugs are reading very white or white with blistering, correct this lean to very lean condition by increasing one size on the main jet.

- If the plugs are reading a tan, brown, chocolate brown or black and fluffy, correct this rich condition by decreasing at least one size on the main jet depending on how dark the plug reads.

- Repeat this procedure until the plug readings are correct.

Fine Tuning

As the above procedure closes in on the correct main jet, it is time to fine tune with the needle.

- Raising the needle clip will lower the needle thus leaning out the AFR mixture.

- Lowering the clip will raise the needle thus enriching the AFR mixture.

If the carburetor does not incorporate a clip on the needle for adjustment, thin washers, if necessary can accomplish the fine-tuning.

After ensuring the spark plugs read correctly, the jetting is as good as good can get and the carburetor will produce maximum power and the most mileage relative to power for the engine and carburetor configuration.

One last procedure is to check the molded plastic inlet fuel elbow for cracks or leaks. A gasoline leak down onto a hot H-D engine bottom end is not conducive to safe riding or motorcycle preservation. This stock problematic elbow is present on predecessor 1976 to 1988 butterfly type Kehein carburetors and early CV Kehein carburetors. There are

aftermarket metal elbows for safe replacement purposes. MidUSA (www.mid-usa.com) has ¼" I.D. (inner diameter)) hose brass fitting, part number 82368 and a I.D. polished stainless steel fitting, part number 82368.

## Conclusion

The ideal and most emissions friendly combustion mixture known as Stoichiometric AFR of 14.7:1 may seem like a new concept to Harley riders because it finally landed on their doorstep in 2006 with the Dyna models. While it is creating much problematic heat issues with the new Harley-Davidson Delphi closed loop EFI, beginning in the same year with the same models, this AFR is very close to ideal for cruising situations also. Cruise situations like around a 14.5:1 AFR so Stoichiometric is not that bad a scenario except when other conditions take the AFR above a super-lean 15:1 and beyond.

Stoichiometric AFR is not new at all, but almost three decades old, when cars became subject to it in the early nineteen-eighties.

EFI became a driving force with automobiles in the late nineteen-eighties, so there was a period of about five years where carburetors had to and did meet air-fuel ratios of 14.7:1 during hot idle and cruise situations. Therefore, it is possible for carburetors to meet this stringent AFR mandate, which is why the reader will see some aftermarket engines with carburetion meeting the current EPA and CARB mandates. Carburetors on cars had an air bypass valve in their intake manifolds. An O2 sensor fed exhaust combusted AFR information to a computer that leaned the mixture up to a 14.7:1 ratio from the carburetor creating the first feedback or closed loop system. However, carburetors had difficulty reaching stricter open loop cold idle and accelerating air-fuel ratios that are less or richer than 14.7:1 particularly with temperature and altitude changes. Optimal performance acceleration AFR is around 13:1. In addition, this is where the problem with emissions controls lies since they dictate leaner ratios than this. These stricter open loop air-fuel ratios in the face of ever-changing barometric conditions and not the

Stoichiometric AFR accomplished with increasing numbers of small vacuum sensors, valves, and hoses supplementing carburetor function. The complexity of all this hardware in combination with O2 sensors and computerization soon made the previously formidable leap to electronic fuel injection a small step.

EFI became a natural progression to more efficient and controllable air and fuel delivery in specific and consistent ratios with the computer fuel maps eliminating tricky altitude and barometric influences. The automotive industry figured this out more than two decades ago delivering efficient cruise AFR's between 14.5:1 and 15.0:1 and more power producing lower acceleration mixtures…just not as low as the performance mechanic likes.

## Tuning the S&S Super E and G Carburetor

The all-time most popular carburetor has to be the S&S Super E and its big brother the G. Both carburetors succeed the wildly popular S&S Super B, which was in use for many years before the E and G introduction circa 1990. I still see many bikes in my service department at Heavy Duty Cycles in Toronto chugging in with a Super B dumping in the gas.

The Super E and G pretty much fixed all the B complaints such as introducing an accelerator pump. Another major design feature of the Shorty E and G is that they allow lots of legroom whereas the B was always in the way. The ease of E and G tuning is a big feature that anyone familiar with the B series will attest. These newer generation carburetors are reliable and easier to adjust than a slide carburetor like the Mikuni Smoothbore.

Fitment and Size

I personally do not like fitting the Super E on older smaller engines like the 1200cc Shovelhead and the Ironhead Sportsters. I also think it too big for the Evolution 883 Sportster although I do favor it on the 1200cc V2 XL. This happening engine can take what the E provides and use the fuel wisely. The S&S Super E works well on Shovelhead 1340cc engines, Evo Big Twins and of course, the TC88 and the upgrade bigger bore TC95.

The TC95 is where one might consider use of the monster G carburetor, but again in my opinion, only if entertaining higher rpm riding. This is a transition area with riding goals and complimentary support components deciding on which carburetor to fit, the E or the G. One way to delineate their use generally is to think of the Super E for throttle response while the Super G might be for peak power.

The S&S Super E has a 1 ⅞” or 1.875” or 47.6mm throat bore and a venturi that measures 1 or 1.563” or 39.6mm. The Super D is the king, more on this later. The Super G earns its reputation for big inch motors with a massive 2 or 2.063” or 52.3mm throat bore and the venturi size of 1 ¾” or 1.750” or 44.5mm.

I could compare these sizes with the Mikuni Smoothbore and the CV Kehein but it is deceptive since these carburetors have variable venturi dimensions that make them more efficient although Mikuni does offer a mega 48mm Smoothbore.

The Venturi and Fuel Circuits

The main criticism I have for butterfly type carburetors is that they are fixed-venturi. Fixed-venturi carburetors like S&S require a pronounced taper as the bore approaches the venturi

further in the throat. This can create a bottleneck at the venturi, which can only correct with a smoothbore throat not available with a butterfly throttle.

The narrower venturi relative to the carburetor throat causes the air to speed up via the physical law Bernoulli's Principle that states a gas or fluid must move through a confining area in the same time-period because of weight and pressure coming from behind. Therefore, the gas or fluid must speed up to accomplish traversing a narrowing to move the same volume in the same time-period. The air-speed increase causes a low-pressure or vacuum area that pulls in fuel from the discharge tube for the main jet that supplies gas for anywhere from 2500 rpm up to redline. The fuel drawn up by low-pressure condition because of higher velocity air travel atomizes with the air stream for delivery into the combustion chamber.

- Fuel for idle and midrange rpm supplies from the carburetor throat near the air cleaner for lower rpm servicing.

- The idle circuit receives its fuel from the midrange intermediate jet.

- The internal idle air-fuel mixture controls by a mixture screw that should not confuse with the external idle-speed screw.

- The external idle-speed screw locates on the left side of the carburetor body.

- Both the intermediate and main jets are non-adjustable via a screw but size in orifice (hole), gradients of a few thousandths-of-an-inches up for more fuel or richer AFR or down for less fuel or a leaner AFR.

- The choke is an adjustable enrichener system that does away with a disruptive and inefficient carburetor throat blocking butterfly-plate. It activates by manually pulling up a lever behind the air cleaner on top of the carburetor body.

The S&S Super E and B require an AFR of about 14:1 for cruise speeds, which contrasts greatly with an optimal 14.5:1 AFR for Delphi fuel injection. The AFR mixture requirements for idle cold starting and acceleration load will drop much lower to 13:1 or slightly lower.

To permit an adjustment of transfer from intermediate range to the higher main jet range controls by an air jet bleed. In previous years before the Harley aftermarket boom, the choice of exhaust systems was limited so the air jet bleed could be constant and fixed at .040”. With the current choice

of a multitude of different exhaust systems and exhaust technology or no technology as the case may be, a need in adjustability for the air bleed became necessary. Larger air bleeds became available on the Shorty's circa 2004. For example, if using drag pipes or other exhausts that have irregular flow characteristics, you know, the ones that are too long, too short, or too fat try a bigger air bleed. Try up to a .060" air bleed with drag pipes and work from there. A larger air bleed delays the transfer from the intermediate circuit to the main jet circuit to a higher rpm. This allows correction of a too rich AFR in the intermediate circuit, which causes a loss of optimal torque production.

The accelerator pump helps correct a negative condition caused by our friend Bernoulli's Principle.

Negative air-pressure allows carburetors to function with

the venturi effect but when opening the throttle suddenly and fully, the airflow will slow because the opening increases dramatically. Remember, a venturi decreases opening size to speed up airflow. Increasing opening size slows the air down because the same amount of air can move lazily through in the same time. Less fuel draws up from the main jet circuit thus delivering less fuel to the combustion chamber causing a decrease in power that reflects as a hesitation in throttle response.

This stumble in acceleration corrects by the accelerator pump that squirts a stream of fuel in with sudden throttle opening. The accelerator pump is adjustable for varying local conditions and engine size. The accelerator pump feed nozzle situates just behind the main jet discharge tube in the venturi towards the manifold but before the throttle butterfly plate.

The Tip of the Day

The beauty and curse of the accelerator pump is that it also assists in cold starts by squirting richer fuel into the engine. The curse is too much fuel will foul the spark plugs preventing start-up. There is no more expert carburetor at fouling spark plugs than the S&S in impatient demanding rider hands.

I always let the enrichener do its choking job without touching the throttle. If the throttle needs to open slightly to assist in start-up, then the enrichener can adjust to provide a little richer air-fuel ratio.

The Super E and G Air Cleaner

These carburetors are at the top of the pile for the older technology and less efficient butterfly throttle types. With this

statement out of the way, I must say their air cleaner design is state of the art. The classic good looks of the teardrop air cleaner is only outdone by internal flow characteristics that allow all the air one could want into the breather through a generous sizing pleated filter.

The inside of the outer cover has a directional air cone appendage that helps direct airflow into the carburetor throat. A large radiused air horn on the inside cover extends the carburetor throat smoothing out the turbulent air into faster velocity laminar flow. The evolving cover design has much the same appearance over decades of use but nothing is further from the truth in terms of efficiency and volume of air delivery. If comparing an E/G air cleaner to that of the B side-by-side, the design evolution becomes readily apparent.

There is nothing short about the E/G cover as it is thicker for a much bigger filter. The filter is now pleated paper or synthetic media and can accommodate filtration of much

more air relative to the B green foam filtration. The E/G outer cover is dimpled for stylish looks but more importantly contains and extends the inside performance directional air cone.

The B inside cover has no air horn and lies flat against the longer B carburetor allowing turbulent airflow into the throat. Of course, this is not such a bad thing because the B has found most use on the hemi-head engines of the day like the Shovel, Pan, and Ironhead that needed all the turbulence they could get to mix air and fuel.

The Super E/G delivers higher volume non-turbulent laminar airflow.

Since the E/G, primary use is to feed the D-shaped Evolution combustion chamber and the Twin Cam bathtub

chamber where the flattop piston squishes the air-fuel mixture turbulently into the smaller combustion chambers very effectively thus mixing the charge into correct proportions for an efficient and high-percentage burn.

## Jetting: Tuning the S&S Shorty

Pre-jetted before delivery, S&S carburetors work well on stock and near-stock Harley's but there is a wide range of use from the 883cc XL up to the Twin Cam 95's 1550cc, so jetting is often necessary if at the extremes of displacement. Again, I do not use a Super E on a small engine like the 883XL, but that is my preference.

Then of course, there are all the elevation-air-density concerns, performance engine modifications, big-inch strokers complicating jetting, and adjustments but the S&S Shorty is simple to tune with the extensive instruction sheets that

provide with every new carburetor.

Starting

Starting adjustments are simple.

1. First, check the setting of the idle mixture screw that locates on the top right side of the carburetor.

2. Turn it all the way in gently until it bottoms and then turn it back out 1¼ turns. Turning in adjustment screws too tightly, thereby ruining their seats, damage or destroy many carburetors. The seats will have a hole in them to allow fuel metering.

What use is a seat, whose metering hole is enlarged thus

increasing fuel flow into a too-rich AFR?

3. Next, gently turn the idle screw on the left side of the throttle body in clockwise until it contacts the carburetor body, and then turn it an additional one-half turn.

This screw and body seating design allows for the extra strain since the additional ½-turn opens the throttle plate just enough for idling.

4. Following this, turn the accelerator pump screw on the right side of the throttle body all the way in until it bottoms.

5. Turn it out two full turns.

Fine Tuning

Fire the bike and let it warm to operating temperatures.

1. Keep the idle at 1000 rpm, either by turning the idle speed screw in to increase idle or turning it out to decrease idle rpm.

2. Do not lower idle to achieve the beloved potatoey-potatoey sound from the exhaust.

Lowering idle speed too much will lower oil-pump-volume flow thus starving the bottom end for sufficient oiling.

3. Now, turn the idle mixture screw in slowly until the engine slows down and begins to run erratically and then back out the screw until the engine runs smoothly.

4. It may be necessary to rest idle speed back to 1000 rpm again as the mixture screw can effect idle rpm.

5. After setting idle mixture and idle speed so the engine runs well at idle, turn to the accelerator pump next.

6. Turn the accelerator pump screw all the way in to eliminate it as a factor for tuning the intermediate and main jets.

The Twin Cam takes a long time to reach operating temperatures, sometimes a good twenty minutes while Evolution models take a little less time and the hemi-head Shovels, Pans, Knucks, and Ironheads warm up the fastest. There is no point in fine-tuning a cold engine because everything will change again at operating temperatures.

- Find a long uphill grade, and roll the throttle on

beginning at about 2000 rpm.

This increases engine load and will determine if intermediate jet sizing is correct.

- If the bike pulls nicely, all is well.

- If sluggish, this indicates a rich AFR.

- Replace the intermediate jet with the next smaller size.

Find the intermediate that locates to the right of the center main jet in the carburetor body. Removal of the float bowl is necessary to access the intermediate and the air bleed to the left of the main jet.

The main jet is easier to change, as float bowl removal is not necessary. Insert a short slot-head screwdriver through the gas drain hole in the float bowl to screw out the existing main jet and screw in the replacement main jet

The left side air bleed is flat into the carburetor body while the center main jet screws into the emulsion tube and the right side intermediate jet is long and thin.

- During the uphill gradient, if the carburetor spits back through into the breather or surges, the intermediate jet is too lean. Replace with the next larger size. Experience will allow the tuner to go more than one size if necessary. In any case, go up or down one size at a time and retest until spitting and surging disappears from a lean condition but does not cause a sluggish rich AFR.

- The reverse situation holds true also, if correcting an

intermediate rich AFR, do not tune downwards too far and create a lean AFR.

- The intermediate will probably be close to a .0295 but do not take this to the bank as there are many variables affecting correct sizing.

- The intermediate jet has the biggest influence on gas mileage.

If getting more mileage is a goal, then go down slightly in intermediate jet sizing.

- If pure performance is the objective then a little richer size is appropriate.

Notice that I said little. Test ride tuning allows a little up-or-down leeway. I always read the spark plugs after a sufficient test ride also. In a Twin Cam white is okay, blistering white is too lean and any coloring is leaning, no pun intended, towards a rich condition.

These readings are a big no-no on a hemi-head where any white is a very lean condition, black and fluffy is a rich AFR and a tan to chocolate brown is the area of choice.

Evolution spark plugs should read between off-white to a tan coloring.

Test riding is superior to plug readings unless the tuner has much experience. Of course, the tuner can use spark plug readings as a first choice and then confirm fine-tuning is correct by test riding.

- Readjust idle after intermediate fine-tuning.

- Main jet tuning is the same as intermediate except at higher rpm and higher speeds of 60 mph and up.

- Generally, main jet sizing will be in the neighborhood of .072 to .076, lower for smaller engines and larger for bigger ones.

- Reset the idle adjustments when satisfied with main jet tuning.

- Finally, return to the accelerator pump that has zero adjustment to allow intermediate and main jet tuning.

The bike is at operating temperature, running but parked.

- Turn out ¼-turn and crank the throttle.

- The engine will stumble. Repeat this process a quarter turn at a time until the stumble disappears.

- If the exhaust emits black smoke, this is an indication that the accelerator pump should adjust back a quarter turn per adjustment until the black smoke disappears but the engine does not stumble when suddenly opening the throttle.

- Black smoke indicates the accelerator pump is squirting too much gas.

Common Tuning Mistakes

1. Jetting too rich is the all-time problem error.

2. The second is to adjust the accelerator pump for too much gas.

3. Attempting to correct an intermediate jetting dilemma with main jet changes is another common mistake.

4. The air bleed should be fine at the stock .040” size unless an inappropriate exhaust system is on the bike. If so, jet up to but probably not beyond .060” or more correctly replace the exhaust system with a performance one.

Do not try to compensate for an inappropriate component by tuning; this is a backwards-inefficient approach.

The S&S Super D

As of this writing, the Super D is the biggest production H-D carburetor in production. Its massive 2 ¼” bore with a stock venturi size of 1.937” and an optional 2.150” venturi saw first introduction way back in 1983. This carburetor is for use on naturally aspirated engines but non-interchangeable variations will also fit with turbochargers, alcohol fuel, and even nitro methane.

Nitro rules the drag strips.

This carburetor needs mega cubic inches with big breathing capabilities although the D has seen use on smaller stroker engines in years gone.

S&S supplies the smaller Super G with its 124 cubic inch Twin Cams.

The S&S Super D is for serious racing.

## Tuning the Mikuni Smoothbore Carburetor

The HSR Mikuni Smoothbore carburetors come in three current sizes, 42mm, 45mm and a giant 48mm.

Mikuni does not make the 40mm Smoothbore anymore but it was a favorite of mine for use on Ironhead Sportster 900cc and 100cc, Evo 883, 1100 and 1200cc XL's as well as the 1966 to 1978 1200cc displacement Shovelhead (74c.i). However, I felt it was too small for larger engines. The problem then was that riders put this too small a carburetor on too big an engine. The problem today is that riders put too big a carburetor on too small an engine. In any case, the 40mm Smoothbore was a welcome addition back in the day for the appropriate sized engines.

The 42mm, 45mm, and 48mm Mikuni Smoothbore

Let me get the sizing out of the way; one millimeter equals .03937 inches. Therefore a 40mm carburetor equals 1.57 inches; 42mm equals 1.65 inches; 45mm equals 1.77 inches and the monster 48mm is a big 1.89 inches.

Compare this with a stock 38mm CV carburetor for Twin Cams and a 40mm sizing for Evolutions. The reader can see how EPA emissions controls and ever-leaner AFR ratios are affecting H-D carburetion. The 1340cc Evo engine uses a bigger carburetor than the 1450cc Twin Cam engine.

- The S&S Super E has a 1 or 1.875” or 47.6mm throat bore and a venturi that measures 1 or 1.563” or 39.6mm.

- The Super G earns its reputation for big inch motors with

a massive 2 or 2.063” or 52.3mm throat bore and the venturi size of 1 ¾” or 1.750” or 44.5mm.

- The massive S&S Super D has a 2 ¼” bore with a stock venture size of 1.937” and an optional 2.150” venturi saw first introduction way back in 1983.

Comparing an S&S butterfly-throttle type carburetor sizing that just dumps the gas in with a more efficient CV or Mikuni slide type carburetor is like comparing apples and oranges. A slide carburetor does more with less in comparison. I will elaborate in the next few pages. I do not demean any of the three carburetors I mention here because they all have their uses and all are at the top-of-the-pile for their specific type.

Mikuni sells these different size carburetors as an individual item without manifold or air breather. This

purchase allows the performance builder to use a higher-flow manifold and air breather of choice.

Mikuni Easy Kits work with original OEM Harley-Davidson manifold, choke cable, air cleaner and throttle cables. Some of them come with a chromed Mikuni air breather cover. HSR 42 Easy Kit fitment is for 1990-1999 Evo Big Twins using the stock CV manifold, 1999-2006 Twin Cams, 1994-2006 1200cc Sportsters and 1994-2002 1200cc Buell's.

While I believe Mikuni to be about the best carburetor around, why use a restrictive stock air breather and not complement the unit with a higher flow manifold? The only rationale reason is to save money but when upgrading, why limit gains with emissions component add-ons on either end although Mikuni supplies an adaptor in order to mount a performance CV style breather.

HSR 42 and HSR 45 Total Kits come with a new performance manifold and a high flow K&N air filter, chrome breather cover and hardware including a blowby breather kit for specific years and models.

- HSR 42 Total Kit fitment is for 1990-1999 Evo Big Twins and includes a breather kit for 1993 and later models and 1999-2006 Twin Cams.

- HSR 45 Total Kits fitment is for the same as the HSR 42 but the Twin Cam kit comes with a MAP sensor location.

- The HSR 48 mm Smoothbore, of course comes by itself since it is special application. Anyone capable of building an engine that it has fitment on will have the knowledge and expertise to mount an appropriate manifold and breather.

These carburetors only fit Harley-Davidson's to 2006 since they are are EFI from 2007 to present. Of course, this does not affect the majority of aftermarket engines and custom applications.

However, Rivera Engineering has kits available for different applications but the basis revolves around the fitment for 124-cubic inch engines that S&S Cycle supplies.

Twin Cam application does need drilling of the MAP (manifold absolute pressure), sensor boss with a drill bit.

Tuning the HSR 48 Smoothbore

Most of what I am about to relay will apply to the other Mikuni Smoothbore 42 and 45 carburetors.

Rivera Engineering (www.riveraengineering.com) sells an

HSR 48 mm kit either as a plain cast unit or as show polished. The kit comes with a high-flow air filter assembly, washable air filter, a chromed billet cover, manifold clamps, seals, and hardware.

So, what does the term smoothbore mean? Smoothbore is a performance feature in that the carburetor bore at the beginning of the throat tapers very slightly as it approaches the venturi. This is where most of the fuel meters in for atomization.

This is the first big advantage of a smoothbore over a fixed-venturi in that there is the potential for much more flow and therefore power if comparing the same size carburetor venturis. The Mikuni Smoothbore venturi size determines by throttle (slide), position. More air flows through at part throttle, which is in the middle of the power band where most

of us do the majority of our riding. More air leads to superior atomization of the introducing fuel and most importantly to a higher percentage cylinder flow. Atomization, while important gets much help inside the combustion chamber with the turbulence created by the rising flattop piston broaching the Twin Cam surround squish band.

The downside of a Mikuni relative to a simpler carburetor like the S&S Super E or G is that tuning may be more complex in difficult areas. However, the tuning produces worthwhile results.

There are four circuit systems in a Smoothbore:

1. Enrichener (choke),

2. Idle,

3. Main,

4. Accelerator pump.

These circuits overlap as in every carburetor. It is better to tune each circuit system separately, especially the main circuit.

Mikuni Smoothbores Tune by Throttle Position not Engine Speed

This is very important. In fact, a worthwhile tuning aid is to wrap masking tape around the throttle grip and make marks for every ¼-turn of the throttle to isolate the circuit that needs correction or adjustment.

Enrichener

The enrichener (choke), on a Mikuni Smoothbore is non-adjustable.

- Ensure the cable has enough free play to allow the plunger to seat fully in the carburetor body.

- Leave the throttle closed when using the enrichener.

- Opening the throttle reduces enrichener efficiency.

Idle Circuit

The idle system controls the air-fuel mixture from closed position idle up to about one-eighth throttle. Terminology gets a little confusing when comparing to other carburetors as the

idle on a Mikuni consists of two adjustable parts, the pilot jet, and the pilot airscrew.

The pilot jet is a fuel jet. The pilot airscrew fine-tunes the pilot jet. Turn right or in will enrich the AFR while turning to the right or out leans the air-fuel ratio. The float bowl must remove in order to increase or decrease pilot-jet-sizing.

Pilot adjustments affect the idle speed. There is an idle speed knob on the left side of the carburetor body.

Turn the idle speed knob out two full turns before fine-tuning it slightly in or out.

The pilot jet is active throughout the rpm range although the main jet supplies the majority of fuel as highway speeds approach.

## Main Circuit

The main circuit system's main influence is from a quarter turn to full throttle. However, it also influences the AFR below quarter throttle.

A needle passes through the main jet. The needle end tapers, but above this, it is straight. The straight end sits in and blocks off most of the main jet from idle up to quarter-throttle.

At quarter-throttle, the needle begins to lift far enough up to allow flow of fuel around it past the main jet into the carburetor throat and venturi. Opening the throttle progressively allows more fuel as the taper decreases in size relative to main jet size.

Different size needles are available but the taper is the same on all of them. Only the straight part increases or decreases in size. This is why the main circuit can influence the idle circuit as the straight part of the needle may allow gas by if it is too small. However, the needle can raise or lower by a retainer clip that holds it in place. There are gradient grooves for retainer clip adjustment. Raising the needle will enrich the AFR while lowering it will lean the mixture. Therefore, in some cases, a simple needle position adjustment rather than replacement may resolve minor idle problems. To access the needle simply involves removing the top cover.

Accelerator Pump

The Accelerator pump responds to sudden throttle openings by squirting a stream of gas into the venturi. This is

necessary because rapid throttle opening increases venturi size decreasing velocity of airflow this reducing the negative vacuum pressure that draws fuel up past the needle in the main jet.

The accelerator pump, of course, also assists in cold starting.

The accelerator pump has two adjustments plus a nozzle that can replace with a larger richer one or a smaller leaner one if necessary. The nozzle controls the rate at which, the pump delivers its fuel squirt while the adjusters determine when it begins and ends. The float bowl needs removal for accelerator-pump-nozzle exchange. The nozzle opening fits up into the carburetor throat before the main jet. The nozzle squirt opening can adjust by turning it with needle-nose pliers. Point the nozzle opening aiming at the main jet needle. The first adjustment is a screw that determines when the gas squirting begins whereas the second adjustment screw

determines when the squirting will stop.

Fine Tuning

Set the idle at 1000 rpm. If adjusting the idle lower for that good old H-D exhaust note, be forewarned that oil pressure to bottom end bearings may well suffer.

- As the engine heats up, idle will likely rise, adjust down to 1000 rpm.

- The idle mixture screw sets at two-full turns out before fine-tuning.

- Slowly turn the screw in until idle begins to drop. This indicates that the mixture is too rich.

- Therefore, turn the screw out a tad until the idle gains momentum.

- Now, open the throttle rapidly to activate the accelerator pump.

- If the engine stumbles, the setting is too lean; if black smoke comes out the exhaust, the setting is too rich.

Riding the bike will tell whether main jetting is too rich, lean or just right.

- Overly rich will indicate by sluggish acceleration, black smoke coming out the exhaust, exhaust backfiring and will use excess gas.

- Leanness symptoms are backfiring through the

carburetor, surging, and detonation as the engine heats up.

Another test is to ride at the bike to 3550-4500 rpm, open the throttle wide open for a few seconds, then close suddenly to

- If the engine does not respond immediately, the main jet is too big or rich; replace with the next size smaller.

- Repeat the test to determine if a needing a smaller jet again.

- If rpm stumbles, there is a lean condition requiring a bigger main jet.

- Find the happy medium between sluggishness and stumbling to get a crisp roll-on response.

The drag racing technique involves accelerating at full throttle between two fixed points.

Change main jets and repeat until the highest mph reading achieves at the second location.

## The Engine Management System

Harley-Davidson is achieving tremendous strides in managing all aspects of engine running. It must balance the goals of producing power, efficiency, fuel economy and less noxious emissions. Unfortunately, fewer emissions will conflict with maximum power considerations.

### Carburetor integrated Ignitions

The V-Fire III computerized ignition system saw first use in 1983. This ECU fuel supervision and ignition system forms the basis of the present day engine-management-systems a full twelve years before first use of EFI on Big Twin touring bikes in 1995. The present day electronic control module co-ordinates engine management in a far more sophisticated fashion but it all builds on V-Fire III goals.

Early Ignition Systems

Mechanical point's ignitions were in use up until 1978. In late 1978, the introduction of the first electronic ignition utilized centrifugal force activating the points-ignition type automatic advance springs and weights. This maintenance prone mechanical advance system depended on centrifugal force literally flinging weights outward for advanced ignition

timing. The weights hook into an adjustable cam lobe whose advance movement activates a solid-state electronic sensor. Slowing engine rpms lessening centrifugal forces allow weak springs to draw the weights back into there sitting (retarded), position.

Points-ignitions have moving parts while electronic ones are solid state with no moving parts. Early electronic ignitions were a hybrid of the two. From a wear point of view, solid state is surely superior to mechanical points. This first electronic ignition was prone to many problems such as outside frequencies like those generated by air traffic controllers at local airports that interfered with the computer dictates to ignition timing.

The V-Fire II corrected some of these problems. However, electronic ignitions did not begin to gain clear superiority over the points systems until the introduction of the V-Fire III.

The V-Fire III consists of three integrated main components; the rotor and sensor plate, which activates the coil and spark plugs; the computerized microprocessor module (ECU or ECM), and the VOES electrical vacuum switch.

The ECU computer has two main functions.

1. Firstly, it opens and closes the low voltage circuits between the battery and the ignition coil so the coil can produce and discharge high voltage to the spark plugs.

The rotor driven by the rotating flywheel pinion shaft and mating sensor plate initiates this action.

2. Its second function is what is interesting for discussing

the evolution of carburetion; the ECU computes the spark advance and retard ignition timing that sparks the spark plug at the proper time.

VOES: Vacuum Operated Engine Switch

So, what does all this seemingly unrelated stuff have to do with a carburetor? The connection revolves around the VOES switch, which mounts into the intake manifold that monitors engine load (cold idle and acceleration), or no-load (hot idle and cruising). Therefore, vacuum levels in the carburetor venturi and thus the intake manifold are an indicator of engine load or no-load. Since vacuum levels inside the manifold parallels engine load and no-load situations, vacuum becomes an effective tool to match ignition timing to the engine's needs. Under heavy loads, vacuum drops and correspondingly retards ignition timing. The vacuum advance unit (VOES) advances or begins ignition timing earlier in relation to piston travel up the cylinder on the compression

stroke or slows initiation of retard ignition timing whereby spark occurs when the compressing piston is further up the cylinder.

The ECU coordinates the VOES signals to advance and retard ignition-spark-timing in order to produce more power and to increase fuel economy. Timing is necessary because of combustion lag; the distance the flame front must travel across the combustion chamber and of course, piston speed. I will elaborate more on these subjects a little later.

A vacuum hose connects the VOES switch to the carburetor or manifold internal opening or venturi through which gas and air flows. Electrical wiring connects the VOES to the ECM microprocessor. When the bike is under load such as with acceleration, a low vacuum exists in the carburetor venturi. The low vacuum opens the VOES switch in turn notifying the

ECU via an electrical signal.

A high vacuum exists in the venturi when the engine is not under load. The high vacuum closes the VOES and informs the ECM.

The V-Fire III programs with two spark curves to delineate the varying engine loads. The maximum advance ignition-timing curve associates with high carburetor-venturi-vacuum VOES signals to the ECU.

Improving fuel economy associates with fewer emissions as combustion that is more complete occurs.

The low venturi vacuum curve retards ignition timing. Retard spark ignition reduces deadly detonation (spark knock), while providing maximum safe performance under high-load acceleration.

Advancing timing creates more heat and pressure that may instigate detonation's spontaneous ignition. Advance timing creates more power but retard brings back the advance just enough to control detonation.

The VOES switching system allows for the proper ignition spark timing via ignition curves to assist in starting; relaxed low rpm riding, hard acceleration, and cruising. It is much more accurate and efficient than centrifugal force driven springs and weights.

## Advance, Retard Ignition Timing

Why do we need carburetor vacuum conditions to advance or retard ignition timing? Advance and retarded ignition timing describes when the spark plug ignites the air/gas

mixture. It takes time for the gas to burn.

Burning obviously creates heat and the expansion of this heat creates pressure to force the piston down to turn the flywheels, which drives the powertrain to turn the rear wheel. For our purposes, heat and pressure is the same thing and are interchangeable. There is no pressure without heat and no heat without pressure.

Gas burns very slowly at first. The molecules begin to move with warmth and progressively agitate with heat before beginning the actual process of ignition or burning. After this combustion lag process of heating up molecules has occurred, the gas burns at a consistent and known rate.

This is extremely important. The consistent rate that gasoline will burn allows for the timing of full combustion to complete at the most opportune time to drive the piston down

on its power stroke with the maximum possible burning expansion pressure. Timing is all about power maximizing. The chemical soup known as Octane stabilizes gas burning to a stable consistent rate. Furthermore, without Octane, gas may explode causing great damage as opposed to a consistent burn.

The spark plug must spark sooner to allow for the extra time that combustion lag takes. Mechanics refer to igniting the spark sooner as advancing the spark or advanced ignition timing.

There are other reasons to advance or retard the timing in order to spark the plug at the correct time to maximize burn of the gas to create heat and thus the pressure to force, the piston down to create torque.

So where must the piston position be when the ECU ignites the spark plug? Theoretically, the piston should be at the top dead center (TDC), as far up as it can travel, compressing the air/gas mixture at the end of compression stroke and beginning of the subsequent power stroke when the spark

plug fires. This means that ignition timing is at with no advance or retard. Another way of saying this is that advance/ retard is at into the of flywheel rotation required to complete all four strokes of one engine cycle. It takes of engine rotation to complete the initial intake stroke and the subsequent compression stroke since each of the four strokes last each.

However, in real life the spark goes off before this to allow for a combustion-lag-time delay and for the controlled burn to travel across the combustion chamber. In actuality, ignition always occurs in real life before the piston reaches top dead center (BTDC), on the compression stroke. Maximum combustion occurs after top dead center, (ATDC) on the following power stroke. In other words, the spark plug begins ignition before the ascending piston fully compresses the air/ gas mixture, before the piston reaches top dead center. Ignition occurs well before the piston begins descending on the following power stroke.

- The first stroke is the intake where fresh air/gas enters the cylinder via vacuum created by the descending piston.

- The second stroke is compression, where an upwards-moving piston compresses the gas mixture.

- The third stroke is power, where the burning mixture produces pressure to force the piston downwards.

- The fourth and final stroke is the exhaust where the upward moving piston forces spent exhaust gases out the valve into the exhaust port and then the pipes.

The four strokes repeat continuously.

Advance or retard ignition timing measures in degrees of

flywheel rotation, on the compression stroke before top dead center (TDC). Advance means that the spark goes off sooner; i.e. further from top dead centre. In any engine, the time it takes for combustion to occur stays the same.

However, if the engine rpm speed increases so does piston speed travel up-and-down the cylinders. Therefore, combustion must start earlier in the compression cycle to allow the gas to finish burning in time to drive the piston down on the power stroke. Conversely, retard ignition must start a little later as the piston slows down during deceleration.

It is all about finishing combustion at the most opportune time to derive the most torque from the piston on its power stroke. This is the primary reason for ignition timing curves. In simplistic form, an ignition curve defines as number of the

degrees of advance/retard change due to engine rpm increases or decreases. This information plots on a graph that for some inexplicable reason, mechanics name an ignition curve.

Degrees of advance/retard express in degrees of flywheel rotation. For example, on a 1995 Dyna glide, the ignition timing is set at (BTDC), before top dead center on the compression stroke between, 650 and 1500 RPM. This is the advance setting while the retard timing is at or top dead centre, (TDC).

Different models have different settings depending on pollution control requirements, variables such as piston speed, ignition systems, and cylinder bore size. For example, the advance setting on an Evolution Sportster is of flywheel rotation before top dead center, between 1650 to 1950 rpm's with retard setting between and

As engine speed increases combustion must start earlier, which means spark timing must advance. Spark ignition-timing retards when starting the engine because the pistons are moving up and down at a slower rate. As the engine, rpm increases when the motorcycle revs up and takes off; the spark advances because of faster piston speed. Now, when cruising, the spark retards again because piston speed slows down.

Let us use some common sense, as we now know a few basic facts to work with.

How about when heads are dual plugged? Does this effect when the spark plug should go off? Well, yes it does, but why? The time it takes a controlled burn to cross the cylinder reduces because there is a spark plug going off on either side. The time should cut in half with allowances made for combustion lag. Therefore, timing should further retard or move closer to top dead center. In this case, to might be

appropriate but is simplistic because of other mitigating factors.

If we use the Evo 883cc or 1200cc Sportster as an example, dual plugging would decrease advance spark setting to to before top dead center. This calculates by subtracting the allowances from the stock setting.

Stroking means the piston must move further up the cylinder and down the cylinder in the same amount of time. Therefore, depending on how radically the stroke increases, the ignition timing must advance accordingly all other factors being equal, which they rarely are.

How about a bigger bore such as when you convert your 883cc XL to a 1200cc displacement or a Twin Cam from 88-cubic inches (3 ¾" bore), up to 95-cubes (3 ⅞" bore), or a

TC96 from 96-cubic inches (3 ¾” bore), to 103-cubes (3 ⅞” bore)? Well, it takes longer for the flame front to travel across the increased diameter of the cylinder bore. Therefore, timing should advance appropriately depending on the bore increase.

However, the combustion chamber size does not increase, as it will on an 883cc to 1200cc XL conversion since the mechanic will hog out the combustion chamber to maintain a sane compression ratio in the Sportster. Therefore, advance should not increase as much on the bathtub combustion chamber as on the XL hemi chamber.

Now complications and necessary compromises arise since these engine changes are usually done in conjunction with each other. Mechanical life is but a compromise. Retard timing with dual plugged heads must compromise with the advance timing requirements of the big bore cylinder and an increase in stroke. How about adding nitrous oxide, which requires ignition-timing retard? Yes, ignition timing gets to be

a complicated compromise.

In 1999, the Evolution Softail models full retard ignition is or TDC while advance is BTDC between 1050 rpm and 1500 rpm. The 1999 Twin Cam 88 ignition timing is and during hot idle at 1050 rpm. This makes sense since amongst other factors the bore of an Evolution is 3 ½" and the bore of both the TC88 and TC96 is 3 ¾", a full .250" larger in diameter. Therefore, the spark must advance more on the larger cylinder.

The Evolution Big Twin has a 4.250" stroke relative to the more conservative 4.000" on the TC88. If the piston strokes were equal, the spark advance increase for the larger cylinder bore could conceivable increase.

The 2007 Twin Cam 96 ignition timing is also and at a hot

idle of 1050 rpm. I would have thought that the advance on the TC96 should be a little more than the TC88 and in a perfect theoretical world; it would be since both have the same bore size but a stroke difference of three-eighths-of-an-inch. I think the reason lies in the fact the TC96 must meet more stringent emissions controls with closed loop EFI Stoichiometric AFR while the TC88 open loop EFI allows for richer AFR particularly in cruise and hot idle modes. Improper spark ignition timing can cause excess engine heat that may precipitate engine damage.

In addition, all of these factors also co-ordinate with the vacuum levels in the carburetor venturi that relays to the ECM via the VOES switch.

Let me uncomplicate this somewhat since it is desirable from a performance point of view, to advance timing until detonation occurs, then retard it until detonation controls.

# Engine Management Advances with EFI

Performance gains are now possible with a fuel delivery-ignition system not possible with inefficient carburetion. The newer Delphi EFI is light years ahead of the inefficient and slow Magneti-Marelli units in use on the first Twin Cams. Fuel efficiency is at an all time high with emissions at an all time low.

The ESPFI (electronic sequential port fuel injection) follows the increase in sophistication incorporating in the experimental 2006 Dynas that works extremely well. One main new feature is an O2 oxygen sensor. This oxygen feedback sensor locates out of visual way behind each header exhaust pipe.

The sensor measures exhaust gas composition and sends signals back to the electronic control module that in turn signals to the electronic fuel injection to adjust fuel delivery for optimal air/gas mixture. This, of course, is within an EPA emissions guideline especially in cruising closed loop or feedback mode with the Stoichiometric air-fuel-ratio of 14.7:1.

The bandwidth of the H-D O2 oxygen sensor is narrow but allows changes within many naturally aspirated performance parameters. Naturally aspirated refers to carburetion and EFI where air breathing operates within the natural laws of physics like vacuum (negative pressure) or momentum (scavenging). Narrow Band oxygen sensors can cover most performance options because AFR changes do not happen fast.

However, an increase in bandwidth 02 sensor like that in use with some aftermarket self-tuning ignitions will make this ESPFI cook in conjunction with supercharging and

turbocharging.

Boost from forced-aspiration can change cylinder pressures and rpm capabilities very rapidly. Wide bandwidth 02 sensors increase reaction time. They can deal with and relay these fast changes to prevent possible injurious situations.

Electronic fuel injection is very much part of the comprehensive engine management system on all modern Harley-Davidson's.

## Rpm Spark/Advance Settings

Spark ignition timing is all about finishing combustion at the most opportune time to create the most burn and therefore, heat. The heat expansion creates the pressure to drive the piston down on its power stroke to derive the most

twisting torque. Twisting torque turns the powertrain including the rear wheel. Torque x rpm = horsepower.

There are many factors that influence the advance and retard spark ignition timing. Advance/retard timing is all about beginning ignition in time for the optimal complete combustion to occur in order to drive the piston down on its power stroke. Spark advance is always BTDC (before top dead center on the previous compression stroke. Retard spark timing is always between BTDC on the preceding compression stroke and TDC (top dead center). TDC is between the end of the compression stroke and the beginning of the power stroke. It is desirable from a performance point of view; to advance timing until detonation occurs, then retard it slightly until detonation controls. Spark must ignite sooner (advance) on

the ascending compression stroke when piston speeds increase via rpm and/or stroke increases because there will not be enough time to burn the fuel charge otherwise. Bigger bore also needs more spark advance because it takes the flame front a longer time to cross the combustion chamber. Dual plugging requires less advance spark timing and more retard spark timing because the two flame travels take less time to burn the fuel charge. Deceleration needs more retard spark timing (less advance timing), because the piston is moving slower thus allowing more time for complete combustion. It just does not make sense to me that the 883 XL would have the same ignition timing as the 1200cc Sportster, whose bore is a full .498" greater in diameter. I advance timing on the 1200cc conversions we do at Heavy Duty Cycles with success. Perhaps emissions controls are hamstringing Harley-Davidson again. Compression and a host of other factors will further complicate the best decision for appropriate ignition spark timing. Necessary compromises arise since these engine changes are usually done in conjunction with each other.

# EFI: Ideal Fuel Delivery

I cannot make the Twin Cam engine exceptionally fast within its capabilities without addressing electronic fuel injection. Some think EFI translates into Ideal Fuel Delivery.

The year 2006 marks the last year for carburetion with all Harley-Davidson's including the Sportster models. Except for the aftermarket, carburetors will fall by the environmental wayside and become dinosaur technology.

EFI has so many advantages over carburetion. However, the driving force behind its use on Harley's is meeting government emissions controls. The precision of fuel delivery on EFI system lowers harmful emissions and improves fuel economy.

For the sophisticated, it can also do wonders for high-performance goals. Carburetors are finicky and imprecise. They cannot deliver the right amount of fuel and air for all rpm ranges and load demands on the engine. EFI can deliver precise amounts of gas and air at any rpm and at any load.

Many argue that the Harley EFI system produces a slower bike than their carbureted counterparts. Well, it is true that they were! The early Magneti-Marelli EFI systems introduced on selected touring models in 1995 were dogs in the performance end of things. As I proceed, this will become very evident because of:

1. Too-small diameter throttle body,

2. Injectors squirting gas into the wrong place,

3. Air leaks,

4. Too few sensors providing information to the ECU.

EFI Benefits

EFI gradually improved over the years until it gained clear superiority in every facet of use with the introduction of the Delphi system in the 2001 Softail models and the 2002 touring models. Check out a V-Rod if you want to see what EFI is capable of in the speed department while meeting emissions dictates. Just imagine the removal of the emissions restrictions on this, the fastest of all Harley's.

Easy starting is another big EFI feature especially in adverse climatic conditions like cold weather starts; just push the start button without touching the The ECU computer has

already told the injectors to put more fuel in because of the cold start condition. The EFI equipped bike will start immediately while the poor old carbureted one must be choked and the throttle primed just right or there won't be enough gas or there will be too much to fire the engine.

Cold idle on EFI bikes may sound racy and there will be no ka-chugging at prescribed rates of 1500-1600 rpm. Hot idle slows down to 1050 + or – 50 rpm. This is a direct result of more gas being injected until the bike warms up. When warm the idle should settle in at 1000 to 1050 rpm but no lower as the charging system can be affected. Too low idle will also negatively affect oil pressure to the bottom end bearings.

While adding numerous parts unfamiliar with a carbureted system like fuel pumps, solenoids and some sensors, EFI also eliminates the need for a cold start enrichment device, (choke), cable and a gas petcock.

EFI compensates gas delivery for differing altitudes while a carburetor must be readjusted and may need jetting changes. Air density refers to the amount of oxygen in the atmosphere. Gravity is what keeps the atmosphere around the earth. As gravity decreases at higher altitudes there is less-atmosphere. As gravity increases as sea level approaches, the weight of the entire atmosphere compresses it giving the air more density. This of course provides more oxygen to mix with the fuel. Different temperatures also affect air density and therefore oxygenation capabilities. More or less air requires more or less fuel in order to have a proper mix for burning in the combustion chamber.

Carburetors are hard pressed to accommodate air density differentials caused by barometric pressure and altitude without jetting changes. EFI has sensors that measure

barometric pressure and temperature to assist the computer with more information to meter the amount of fuel to inject into the intake ports. More accurate fuel delivery under differing but specific engine loads and operating conditions is the big plus over the relatively wasteful carburetor.

However, one of the problems, as I see it is that new-fangled terminology describing new ways of delivering fuel can be threatening.

Fuel linking Spark Maps or Tables

The spark and fuel maps also known as tables are a biggie here and conceptually intimidating and yet, when explained is disarmingly simple. Fuel and linking spark maps that chart every throttle position combining with load or how hard the engine is working are memorized inside the electronic control module. Each combination of load and throttle position sent

to the ECU via sensors gets its own cell in a fuel/spark map. The fuel requirement for each combining with more sensor information such as manifold vacuum (MAP), also logs into individual linked cells.

This ECU signals the injectors when and how long to open according to the fuel-spark mapping cell positions for load, throttle position, and air-fuel ratio. The linking ignition-timing table or map instructs the advance or retard positioning for the spark plugs to fire. Having the ability to adjust these settings at will for every rpm and load setting is the trick to achieving horsepower.

A carburetor draws and mixes fuel and air via vacuum, manifold pressure differentials and venturi air speed.

EFI Injectors

Electronic Fuel Injection delivers fuel under pressure by a pump to gizmos called injectors. Injectors inject fuel as a pressurized spray. There are two separate fuel injectors in a Harley engine, one for each cylinder. The longer an ejector nozzle holds open will determine the amount of fuel sprayed/ squirted/injected, take your pick, onto the back of the intake valve on Delphi systems just as it opens and into their respective cylinder.

The early Harley Magneti-Marelli EFI systems spray the fuel onto the inside throttle body near the intake ports but this is silly since more efficiencies and speed develop by squirting it directly at the backs of the opening intake valves.

Twelve volts of electricity are sent to each injector as soon as the ignition is turned on and the starter button is pushed in order to start the bike. There is an electromagnet inside each

injector. It grounds when a transistor inside becomes energized thereby opening the injector valve or nozzle.

EFI Injector Pulse Width

The time the injector valve is open is called the injection pulse width. The pulse width gets longer when the ECU requires more fuel. This on time or width measures in milliseconds (thousandths-of-a-second). Idle requires three or four milliseconds of pulse width. The pulse width of the injectors increases providing more fuel as the bikes engine load increases.

This all determines by the ECU, whose computer processor receives sensor information, refers to the memorized fuel and spark tables that select injector pulse width on-time spraying and amount of spark advance or retard for ignition. The ECU

has the ability to change pulse width over a hundred times per second. Compare this with the barometric pressure and vacuum response time of a carburetor that will require full seconds not hundredths-of-a-second.

EFI is a long way from a carburetor when information dictating fuel spray is processed in less than a hundredth of a second and volume is measured in delivery per thousandths of a second. I try not to beat up on carburetors because I like you have affection for them but it is tough to do so when writing comparisons.

EFI Pressurized Fuel Delivery

Fuel is kept under pressure in order for instant availability. Excess gas simply pumps back into the fuel tank. There is no lag time with EFI. Appropriately, the jargon for this is injector on time.

This contrasts with carburetion where atmospheric pressure differences draw in gas that must also vaporize, which is a relatively slow process. Intake manifold pressure is a measurement of the flow of air through the throttle body venturi and intake manifold. Suction from the descending piston on the intake stroke initiates the flow of air through the fuel delivery system. A carburetor creates a partial vacuum to draw fuel into the air.

EFI sensors note changes instantly via a manifold air pressure sensor (MAP) to send signals to the ECU to dictate memorized fuel and spark requirements. Carburetion is comparatively slow and inaccurate relative to the instant precise reaction time in a fuel injected system.

Measuring Engine Load

A major difference between older and newer EFI systems is how engine load measures. Once again, engine load is how hard the engine is working. A steady throttle position createss little or no load while cranking it puts load on the engine and powertrain.

The older 1995-2001 Magneti-Marelli EFI measures load by throttle position only, which is educated guessing. The newer Delphi EFI is much more accurate. It still has a sensor monitoring throttle position, but combines it with measuring manifold pressure, the intake air temperature and the engine temperature of each cylinder along with crank position and speed.

Ion Sensing Detonation: Anti-Knock Sensor

A relatively new sensor that detects spark knock also

known as pinging or detonation monitors engine performance.

Spark knock is a daily fact for many Harley riders.

The anti-knock sensor detects detonation through a procedure called ion H-D ion-sensing does not work well enough to eliminate spark knock on many stock Twin Cams unlike the aftermarket S&S anti-knock sensor that will retard ignition timing until the knock subsides but it is a welcome and necessary addition nonetheless.

Detonation may also control by the addition of cooling gasoline via richening the AFR or retarding ignition timing. Richening AFR may violate EPA and CARB dictates.

Three-Phase Heat Management System

Finally another way attempts to address internal damaging heat from the engine. Harley, between a rock and a hard place, is valiantly attempting to control heat through a 3-phase heat management system that automatically varies the idle speed and fuel delivery. Sometimes riders will think

something is amiss with their bikes idle when the ECU institutes heat management.

1. Phase I occurs when the temperature sensor sends a signal to the ECU when the cylinder head reaches F. In response the ECU will lower idle speed until the temperature drops or the engine reaches 800 rpm.

2. Phase II comes into play if engine temperature reaches F.

The ECU enriches the gas to air ratio to allow some extra cooling.

3. Phase III institutes skip function only when the bike is not moving and the temperature sensor indicates heat above F. There is no combustion with skip function. Fuel injector

pulsing (squirting) temporarily halts.

Air is still drawn into the combustion chamber and expelled through the exhaust. The flowing air-cools down the cylinder temperature. The uninitiated will think that the engine has a problem and is misfiring. Well, I am sorry to report that despite the foregoing attempts that the Twin Cam still runs insufferably hot on a summer day.

H-D makes ongoing attaempts to deal with insufferable heat such as EITMS (engine idle temperature management system), which I cover in detail later in this chapter plus in other areas of this book in Chapter III: H-D 110 Cubic Inch CVO and in Chapter XVI: 2009 Model Changes as well as in Volume I: The Twin Cam.

Cylinder Synchronization

Cylinder synchronization is engineer-speak for coordinating the optimal time for the spark plug to begin igniting of fuel. This of course, occurs as the piston is ascending the cylinder bore on the compression stroke. The ECU computer knows cylinder synchronization at all times through reporting sensor(s) especially the CKP (crankshaft position sensor).

On early Twin Cams, the cam position sensor (CMP also known as CPK) reports the piston location via cam lobe position to the ECU to help control spark ignition. The cam position sensor measures this via a magnet registering a break in the rotating camshaft. Every time the magnet in the sensor detects this break it sends a signal to the ECU computer. The ECU then knows the position of the cam lobes, which indicates where the various valves (valve events) are in their respective opening and closing cycles. More importantly, the computer is able to determine from cam location and thus the

valve events the piston positioning in their respective strokes so the spark can begin ignition at (retard spark timing) or before (advance spark timing) TDC (top dead centre). It is only necessary for the sensor to detect where one rear cam lobe is since the other is fixed on the same shaft. Furthermore, the front cam location with its two lobes fixes to the rear cam via the secondary timing chain.

The Magneti-Marelli electronic fuel injection system in use on 1999-2001 Twin Cams utilizes the CMP, cam position sensor, (32793-00, 32496-99), that identifies the cylinders and their respective pistons to aid the computer to synchronize spark firing. In our experience the CMP sensor has proven to be problematic on these early models that coincidentally have the self-destructing early rear cam support ball bearing. One problem leads to another and it is fair to say that in this case the CMP sensor becomes the victim if the rear cam supporting ball bearing begins to allow increased cam

sideway play in the initial stages of bearing breakdown. I have written extensively on this early Twin Cam timing chest problem that was corrected by the replacement of the ball bearing with a roller bearing to support the rear cam in late 2000 in Volume I: The Twin Cam, Chapter II: 1999 -2000 Teething Problems, and Chapter III: Timing Chest Breakdowns and

The ill-supported rear cam begins to rub against the cam position sensor. The very sensitive power coil in the cam position sensor becomes damaged thereby preventing it from accurately reporting the cam position to the ECU. Furthermore, the newer and more sophisticated Delphi EFI engine management system in use on Twin Cams from 2001 to present mercifully allows the elimination of the cam position sensor in favor of the more reliable crankshaft position sensor.

Each year brings improvements and the 2006 Dynas are

leading edge relative to what went before.

## EFI Cylinder Synchronization

Cylinder synchronization detects on later Delphi equipped models by minutely measuring the slowing of crankshaft rotation on compression strokes. As the piston is ascending the cylinder on this cycle or stroke it compresses the air and gas into a smaller and smaller area into the combustion chamber. The resistance to this compression of gases is what causes the crank to slow down very slightly.

The crankshaft position sensor magnet that locates in the front left side of the crankcase measures the rotational slowdown. It magnetically reads the 30-breaks (teeth), in the rotating steel mass known as the left flywheel of the crankshaft. If visually observing a cutaway engine, the human

eye would not be able to distinguish any slowing in movement caused by the increasing compression of gases. Computerization easily reads this infinitesimal change in rotating speed to accurately gauge when to initiate spark.

This is a great improvement over previous electronic ignitions that did not have the luxury of sensors sending precise information thousands of times per second. The advantages over mechanical points/condenser antique methodology in use until 1978 are almost too great to measure. Points can be fun or extremely annoying depending on your perspective but I think their popularity along with the soon to be antique carburetion derives from those that need to be in charge and to control.

Computers move too fast for those that have control issues and many times can diagnose and sometimes fix their own malfunctions.

I will be the first to admit the deep satisfaction that derives from tinkering with a broken down motorcycle and to hear it fire up and roar in appreciation because of my actions.

However we are now in the age of self-tuning ignitions. The aftermarket has developed them already and Harley surely will incorporate this function in future years. S&S was the first to do this on a mass scale with their proprietary intelligent spark technology for carbureted models but requires skilled technicians to set it up. Terminal Velocity is another good example of self-tuning for EFI and does not require new skill-sets.

## Sensors

The stock Harley ECU has fixed values in order to comply

with EPA dictates. The system design runs extremely hot in order to burn a higher percentage of the fuel to lower harmful emissions. The problem would resolve within the parameters of a 2-cylinder V-Twin configuration if the system's ignition timing component was adjustable or self-adjusting in order for the timing to retard when pinging occurs.

Another way that the aftermarket deals with this is by richening the system, which is raising the ratio of gas to air, in order to cool off the heat, so to speak. The truly good systems incorporate both a richening component and a spark modifier although spark modification is not as big an issue with the Twin Cam as it is with Evolution models. Ergo, a whole industry is developing over modifying first the Evo EFI engine management system and now the Twin Cam. It develops constantly and mostly becomes user-friendlier all the time.

Before we get into these modifiers let's look at one way to

make the engine run a little cooler by relocating the air temperature sensor away from the fuel injection throttle body.

## Relocating the Air Temperature Sensor for More Gas and Less Heat

An American Iron Magazine reader complained that his stock 1450cc FLHRI engine pings while under load, especially with two riding up even though using 93 octane fuels. H-D dealers in analyzing his complaint conclude that everything is within accepted specifications. I replied that could very well be true.

Confidentially a Factory representative explained that due to EPA restrictions, the air temperature sensor (ATS), located in the throttle body would lean out the mixture as air temperature increases. More heat equals the ability to burn a

leaner mixture of gas, which has a higher air to gas ratio. Sounds good, except that the situation will start to feed upon itself; the ECU leans out the mixture as a result of receiving signals from the air temperature sensor. Now, the engine will run hotter causing the computer to lean out the mixture further because, again, of signals sent by the temperature sensor.

The Factory representative recommends, and I heartily agree, installing a second air temperature sensor in the back of the air cleaner mounting assembly. Locate it so that the temperature-sensing probe is inside the confines of the air cleaner element. Unplug the lead from the stock location air temperature sensor and plug it into the newly installed sensor inside the air cleaner element. Leave the stock sensor in place, as its new function is to plug the hole.

The new sensor will now send back cooler signals to the computer, which in turn will instruct the injectors to increase pulse width to enrich the fuel mixture. This equates to cooler, which will at least lessen and may eliminate the pinging. The thinking mechanic learns to fool the computer with sensor finagling to fix stock factory problems and/or to increase performance.

Another sensor, the bank angle sensor turns the bike off if the bike tilts over more than 45-degrees for one second while running. This protects the engine in a spill. It also replaces a manual kill switch for an automatic one for the drag racers in the crowd.

The bad news from a performance perspective is that your Harley utilizes the crankshaft position CKP sensor to determine ignition timing and it is non-adjustable. Therefore, the Factory Twin Cam ignition cannot mechanically adjust, which drives older mechanics nuts…and me too until I

adapted.

I may be a dinosaur but I am an

The mechanic cannot time a Twin Cam in the traditional way by locating a front or rear cylinder timing mark on the left flywheel by removing a timing hole plug to peer into the engine cavity. Don't bother looking for this timing plug in your Twin Cam, as it is no longer there. In the good old days, once the timing mark was found, the sensor plate also known as a pick-up unit or the point's plate in pre-1977 engines inside the timing inspection cover on the engine right side nosecone could rotate to adjust the timing. There is nothing in this compartment anymore. The empty cavity allows access to the cam position sensor on the early Twin Cams but this component became obsolete circa 2001-2002. Methinks that H-D does not want to change the appearance of the engine by

designing out this cavity since the Harley rider is skittish about styling changes, functional or not, to our beloved V-Twin engine.

Now some aftermarket ignition/fuel modifiers can set up the initial timing and the ignition timing advance slope. Furthermore the engine may tune manually by knobs or by interfacing a computer with an interface cable attaching to a laptop to custom program a timing advance curve.

The Various Sensors

The new Twin Cam times by a singular crank position sensor (CKP).

- The crank position sensor signals the computer piston location and engine RPM.

- Air density measures via an intake air temperature (IAT) sensor in determining oxygen content that will affect fuel richness or leanness.

- The vehicle speed sensor (VSS) signals to the ECU how fast the bike is traveling.

- The VSS sensor also sends electrical pulses to the speedometer module. The module can turn the signals off after a certain distance indicated by the VSS.This is why turn signals will stay on at rest and turn off when moving. A faulty VSS sensor may result in turn signals staying on and not turning off.

- A MAP (manifold absolute pressure), sensor informs the ECU of engine load if any the engine is under via manifold vacuum. It measures the vacuum in the manifold throat

between the throttle body and the injectors. Vacuum engine load or no-load situations indicate by pressure (vacuum) levels.

- TMAP is the temperature manifold absolute pressure sensor.

- An engine temperature (ET), sensor also helps the ECU computer control fuel mixture because the hotter it is the leaner the mixture will be.

- The TP or TPS sensor indicates throttle position.

- TGS is the new acronym for throttle grip sensor.

- BAS is the already discussed the Bank Angle Sensor along with the AKS sensor.

- AKS is the Anti-Knock (Ion), Sensor earlier.

All the sensors excepting the VSS speedometer sensor, which helps determine idle speed control, signal information allow the ECU to coordinate air and engine temperature conditions, engine rpm, oxygen levels in the air and engine load to set fuel requirements and minor changes to ignition timing via preset cells in the spark map. All these coordinates indicate constantly changing specific cells on the spark map in the computer processor to dictate fuel and ignition settings.

The H-D scanalyzer tool reads all the engine data in order to diagnose breakdowns also shows timing on both cylinders while the ECU advances and retards timing 2 to 3-degrees independently on each cylinder on newer EFI engine management systems.

To my knowledge, Competition Cams (www.competitioncams.com) marketed the first EFI engine management system fuel modifier that I used back in the nineties. It was a godsend for the time but had tuning difficulties with some performance modifications. The original units are ancient technology now.

Power Commander was the first sophisticated fuel modifier and its latest version also modifies spark timing. Power Commander is an excellent system but is for people that know what they are doing. I have used them since inception with great results for the increasing technology of the times.

Expertise for modifying H-D ignitions is improving almost monthly so a good understanding of how these modifiers work is necessary for mechanics to get the most out of them. A guy

like me will start writing articles about say Power Commander, which requires expertise and readers will go to their dealer or aftermarket shop and say, “I want a Power Commander”. The customer doesn’t query their skills for this type of work. If the shop doesn’t have a computer out back for downloading spark maps from the Internet site to the Power Commander or a dynamometer to tune them then what’s the point? Some shop/dealers don’t want to lose business so they install the unit and then the problems begin because the installers don’t have a clue. The customer gets ripped because he or she now has to pay twice. This supposedly feel-good experience can soon turn into a tuning nightmare.

Power Commander offers an expanding big pile of different spark maps tailored to fix inadequate delivery of fuel, wrong spark timing or simply eliminating an irritating hesitation at a certain rpm in acceleration or cruise mode. The ever-evolving and sophisticated Power Commander will accommodate whatever fuel or ignition need or desire from

correcting hiccups to most hi-performance changes. A multitude of maps are available online for hundreds of different performance modifications. Plugging in a laptop computer into the Power Commander will allow a knowledgeable technician to create a new spark map or maybe just correcting a few cells in the existing map.

Alternatively, Power Commander has dynamometer Tuning Centers with software that automatically remaps your system. In addition to the dyno and software is an exhaust gas analyzer, which shows the technician the exact air-fuel ratio on a dyno run. Furthermore an Eddy Current Power Absorption System in conjunction with the dyno can run the bike at any rpm and load for precise tuning.

Contrast this with Harley's initial burn download methodology, which is restrictive and only affects very

specific changes. The problem with a fixed download is that it may not cover the modifications the rider wants. Harley-Davidson then met this challenge/inadequacy with first, the Race Fueler, a fuel modifier and the sophisticated and not always user-friendly SERT or Screamin' Eagle Race Tuner, a mapping programming device.

The DFO, Digital Fuel Optimizer, also known, as Techlusion is a current practical and user-friendly gas modifier that is excellent with air breather and exhaust changes but cam changes may confuse this device. Early fuel modifiers are throttle based while the DFO is load based so that it responds by adding fuel quickly with full throttle acceleration under load.

S&S has an excellent carbureted system called Intelligent Spark Technology (IST) that has a knock sensor to detect detonation. The system automatically adjusts ignition timing to eliminate the deadly pinging or spark knock.

Another current favorite of mine is the Terminal Velocity fuel management computer since it incorporates an EFI 02 sensor and tunes itself. The Terminal Velocity easily installs but no tuning is necessary. Simply go out for a ride. The rider can actually feel the bike tuning itself while on a test ride after installation; after 50 miles it totally dials in.

This is the new way both for H-D and the aftermarket by utilizing O2 oxygen

sensor(s) that analyzes the exhaust gases in the pipe(s). Exhaust analysis with anti spark knock capability in self-programming ignitions are now the way to go that modify both ignition timing and fuel requirements.

EFI is the way of both the present and the future. EFI

becomes easy once you get over all the confusing acronyms and technical jargon by beginning to understand the basic pressurized injection principles.

The Villain

The AFR going above Stoichiometric 14.7:1 is the true villain.

We can lower temperatures after the fact with assists such as EITMS (engine idle temperature management system), cylinder cooling fans, synthetic oil, larger capacity oil tanks and dual 10-core Jagg oil coolers.

However, the cure will revolve around one or a combination of two things; utilization of components that can withstand and function within higher operating temperatures and the second is to lower temperatures. If we can keep the

rear cylinder at Stoichiometric AFR, we can probably live with this and obey closed loop emissions dictates.

Lowering slightly below Stoichiometric in cruise mode and much below in acceleration open loop mode will give increased engine longevity along with performance improvements.

Two Sub-Types of EFI: Open and Closed Loop

There are two main types of electronic fuel injection to receive the various sensor readings such as ambient temperature, engine temperature, throttle position; manifold air pressure (vacuum), crank position and others in order for the memorized fuel-spark tables to respond.

Once a stock component part replaces with another like an

upgraded exhaust, high flow air cleaner or longer duration-high lift cams; these changes will disturb the factory-installed air/fuel calibration settings that closely link with ignition tables in the electronic control module (ECM). In order for your Harley to run properly with a change or changes, adjustments must be made to give the ECU necessary instructions of how much air and fuel are required.

This is not so simple. Carburetors require mechanical jetting changes, which is easy. EFI needs electronic adjustments fed either into the ECM or modifying signals after they leave the ECM. The ECM fuel and coupled ignition tables are fixed.

Up until very recently from 1995 to 2006, Harley-Davidson as well as most of the aftermarket chose open loop EFI for the ECU to receive, interpret and control fuel settings from sensor readings. The vast majority of aftermarket ECU modification systems to alter fuel settings are very limited to

specific changes without further adjustment. Earlier units cannot adjust without prompting. Therefore anytime a change is made such as an exhaust, neither the Factory nor the aftermarket calibration tables are any longer valid. Sometimes this will be injurious to the engine if a lean condition develops. Other times the fuel will become too rich and wasteful.

It is possible to rewrite the stock ECM calibration tables but hardly practical. Most aftermarket units allow a choice of settings or modified tables for specific changes but they need changing every time another modification is made.

The automotive industry has been using closed loop EFI forever favoring it over open loop EFI. Harley-Davidson finally made the switch over to closed loop operational fuel control in 2006 with the Dyna series. It carries over into the

full 2007 lineup including Sportsters. The reason is the addition of one of the now more important sensors, the O2 or oxygen sensor. This sensor monitors oxygen content in the exhaust stream out the exhaust system. It is real time monitoring and diagnosis of the air-fuel-ratio's efficiency. Now the fuel delivery can correct automatically to the Factory calibration tables basing on EPA dictates.

Whether one agrees with the preset calibration tables or not, the point is that they become accurate in the predetermined and thus correct air-fuel-ratio. Many aftermarket exhaust companies have begun the switchover by supplying their exhausts with threaded openings known as bungs for optional oxygen sensors. This allows easy installation of O2 sensor(s).

Now self-tuning ignitions like the Terminal Velocity can

now recalibrate over 500-times-per-second to adjust to not only any changes the rider may make like an air cleaner but to constantly changing load conditions that occur while riding like adding a passenger or trailer, going up hills, going down hills, accelerating, decelerating or simply cruising. The ignition no longer cares as it simply reads the exhaust stream, analyzes the parts-per-million oxygen content, and delivers constant information in milliseconds to the ECU so that a decision will also be made, again in milliseconds to determine fuel requirements instantly. It will also compensate for altitude changes lowering fuel as elevation increases because of less available air and increasing fuel as elevation approaches sea level for the same reason of now more air.

This is new to the motorcycle industry but again, not the automotive especially in racing circles. Some pretty big names use this technology such as BMW, Mercedes, Honda and the domestic Big Three. Furthermore, Formula One racing also uses this expertise.

Closed Loop Electronic Fuel Injection

A closed loop EFI requires oxygen sensor(s) to feed exhaust composition Lambda 1 (14.7:1 AFR), to the electronic control module. Closed loop EFI functions at Stoichiometric air fuel ratio of 14.7:1. Closed loop sometimes referred to a feedback EFI needs a full circle to reach its mandated AFR. The full circle ends with the ECU referring memorized pulse width to the fuel injectors, which as we now know is how long they

stay on for. This, of course, decides the amount of fuel injected.

The engine goes through its four strokes until completing the exhaust cycle. The O2 sensors measure Lambda and send a signal back to the ECM (ECU). The ECM will increase, or more likely in the real world decrease fuel injector pulse width to reach a Lambda of one (1), which is equivalent to Stoichiometric AFR of 14.7:1. Closed loop EFI functions in cruising situations where fuel requirements bend toward a regulated Stoichiometric 14.7:1 AFR as well as hot idle. Open loop functions for accelerating and cold idling particularly during engine warming after start-up.

Open Loop EFI

It is unlikely Open loop EFI will reach an AFR of 14.7:1.

However, the likelihood is a richer condition for acceleration of around 13.0:1 AFR with a Lambda of less than one (13.0:1 ÷ 14.7:1 = .884 Lambda), will exist with open loop. An air fuel ratio of 12.0:1 (12.0:1 ÷ 14.7:1 = .816 Lambda) is the H-D engine default rich idle mixture.

When the computer ECU, decides the engine has warmed sufficiently it will put the cold idle open loop mode into hot idle closed loop mode and seek a 14.7:1 air fuel ratio. Blipping the throttle or otherwise taking the engine past 1200 rpm from idle of 1000 + or – 50 rpm will engage the closed loop mode and heat up the engine further.

Open loop EFI is not EPA functional for the cruising air fuel mixture ratios on the newer bikes when closed loop 14.7:1 AFR comes into effect. Accelerating and rich idling cannot use the lean AFR used in Stoichiometric closed loop EFI because the engine will stall at idle and stumble while accelerating generating even hotter injurious cylinder

temperatures. These operating conditions require a richer AFR.

In closing, our current new Harley's in 2006 and later use open loop EFI during start-up with initial rich idling. Once engine temperature warms up the idle, the ECM will enter closed loop mode and seek Stoichiometric 14.7:1 AFR. Acceleration requires open loop EFI from closed loop cruise mode and for passing because of a richer AFR requirement. The ECU utilizes closed loop EFI while cruising to meet EPA mandates. It is not an ideal situation for an air-cooled engine.

Delphi EFI Ongoing Improvements

The previous angled fuel injectors are now at more proper 25-degree angles.

The 2006 Dyna TC88 introduced a Delphi 46mm EFI venturi throttle body.

The 2007 TC96 models carried over the use of 46mm bodies with a 50mm Screamin' Eagle option. Then the 2007 CVO 110 also fits with the 46mm throttle body…duhhh! The Screamin' Eagle upgrade is now a 58mm body, which in my opinion is making a mistake going the other way to too big from too small. The S.E. Pro High Flow 58mm EFI throttle is 8mm bigger than the previous high performance option of 50mm, a 16% increase. The part number 27713-08 is for 2008 touring models with fly-by-wire throttle. In other words it accommodates the slave servomotor that electronically responds when turning the throttle. This is called electronic Throttle Control technology. Of course this 58mm body will not go to waste, as it will be ideal on a larger engine than the CVO 110.

JIMS now provides both a 58mm throttle body and

manifold along with a massive 62mm throttle body. They have an elliptical intake tract to enhance horsepower and torque numbers throughout the rpm range.

Just to demonstrate how far this is advancing, this all pales in comparison with the anemic and paltry 38mm throttle body used on the first Magneti-Marelli electronic fuel injection systems used on Harley-Davidson 80ci Evolutions and the first TC88's.

Delphi EFI Temperature Cooling Assist

The EITMS (engine idle temperature management system) also known as idle temperature cool down turns off the rear cylinder fuel injector at hot idle. Why would we do this when gas is the cooling agent and air causes the villainous heat? Burning gas assisted by maximum air generates the heat from

too high air fuel ratios.

In this case the Factory uses air as a cooling agent because there is no ignition of gas in the rear cylinder during EITMS. The 2007 engine idle temperature management system in simple terms offers a cooling feature for hot idle conditions. This engine calibration procedure is available for all FLHT touring models plus police specific FLHTP, FLHP, and FLHPE motorcycles.

This is an option available for all these models free of charge from your H-D dealer. This meets the needs for those that ride in very congested traffic on a regular basis or need to leave their bikes idling as in a police situation. The calibration offers limited rear cylinder cooling when cylinder temperatures reach a certain level. Once EITMS activates, the rear cylinder still functions without fuel. Air still sucks in on the intake stroke, compressed on the compression stroke, decompressed on the power stroke, and exhaled on the

exhaust stroke. The air acts to cool the engine rear cylinder while only running on the front cylinder.

Once activated the EITMS will only function when there is high engine temperature, the engine is at idle speed, the bike is not moving, and the transmission is in neutral or if in gear with the clutch lever pulled in thereby disengaging the clutch. When all of the above conditions meet, the rear cylinder will stop firing. EITMS will allow renew supply of fuel to the rear cylinder if temperature decreases, the throttle is increased, the bike moves forward, the clutch engages or the bike shifts out of neutral with the clutch engaged then the rear cylinder and thus the engine will begin running normally.

The potato-potato idle sound will be different since only one cylinder is contributing one potato to the noise rhythm while there is the sound of the whooshing of air from the

other. This will also affect the exhaust smell. I mention the above to allay fears that not all is right if not prepared for these EITMS normal symptoms.

This free one-time activation and one-time deactivation if you find it unnecessary or you simply dislike the feature is available for the models listed above. Quote Service Bulletin M-1191; labor code 8202; customer concern code 9203 and condition code 9110 although referring to M-1191 will do the trick.

Please see Chapter XVI: 2009 Model Changes for further EITMS updates.

What is Honda Doing?

Take a look at what Honda is doing. It is developing EFI even for its smallest 50cc motorcycle engines. As progressive

as this is, Honda looks to the day when a fuel delivery system will be obsolete.

Honda also is reportedly developing an internal combustion engine that has no harmful emissions. Furthermore, it is currently beginning to utilize hydrogen fuel cells to power some of its bikes. This eliminates the internal combustion engine altogether. Hydrogen is used to power the bike and the only exhaust is drinkable water!

Then there are electrically powered motorcycles and on it goes.

## Performance EFI Ignition and Fuel Modifiers

In review, an electronic control sometimes called an electronic control unit module (ECU or ECM) has tables also

known a fuel maps or a spark matrix. The tables or map or matrix, take your pick have predefined and fixed fuel settings for rpm and throttle position (load). Hence the term, map matrix, or table. I will intersperse this terminology but they mean the same thing as the spark or ignition component ties in with the fuel settings.

Fuel and Ignition Maps in the Electronic Control Module

Visualize a map as a multi-cell table where each cell represents a certain rpm and throttle or load setting to correspond with the rider's twisting of the throttle. Whether, the bike is under load or not determines by acceleration or load or cruising, which is a no-load situation. The corresponding ignition curve component linking to the fuel settings will advance spark ignition or retard spark ignition.

The tie-in is simple to understand. For example, faster acceleration requires advancing the spark. The piston moves faster up the cylinder so the spark plug must begin ignition earlier in the upwards compression stroke to burn the highest percentage of gas at the optimal time so the expanding heat will drive the piston down on its power stroke.

The Octane in the gasoline creates an even burn situation so that there is predictability to how long it takes for the flame travel to move across the combustion chamber.

If decelerating, spark will retard, igniting further up the cylinder on the compression stroke because there will be more time for the consistent timing of the flame front to accomplish its goal of best possible combustion in time to drive the piston down to create torque via the turning flywheels, which twists (turns), the sprocket shaft.

In cruise situations the spark table again ties into the cells that have little or no load with modest fuel requirements approaching Stoichiometric and will set spark ignition somewhere in the middle of advance (sooner) or retard (later) spark timing.

The ECU depends completely on information provided by electrical pulses delivering from sensors. The MAP or manifold absolute pressure sensor sends pulses to the ECU indicating vacuum conditions in the engine manifold. This indicates load or how hard the engine is working. The throttle position sensor tells the ECU if it is twisting indicating acceleration loading or constant indicating a cruise no-load situation and everything in between for the succession of map/table cells. More sophisticated electronic control modules will have an anti-knock sensor to indicate when detonation, the great engine destroyer, is occurring.

The ECU may advance ignition timing in a no-spark-knock situation or retard spark timing in a knock or detonation condition.

The sensor(s) in the exhaust pipe header(s) is a very welcome addition with first use in the 2006 Dyna models and all Twin Cams beginning in 2007. It is important to note that the ECU only pays attention to information in closed loop mode and ignores it completely in open loop modes of idle and full throttle acceleration. I explain elsewhere that the air-fuel sniffer component to the dynamometer is very necessary to read the air-fuel-ratio of the exhaust gases leaving the engine. Harley-Davidson now has its own onboard sniffers that very accurately tell the ECU the efficiency of combustion to reduce emissions in closed loop mode.

The modern ECU will change fuel delivery to accomplish a

14.7:1 air fuel ratio in closed loop modes, which are hot, idle and cruise situations. This Stoichiometric ratio is the most efficient for approaching complete combustion producing the fewest harmful emissions. At the end of the day, the ECU is efficient at determining spark timing and fuel delivery in a power and emissions compromise that is until the advent of closed loop EFI on the 2006 Dyna models.

Harley-Davidson is having difficulty in coping with the heat generation emanating from the high Stoichiometric AFR of 14.7:1 with an awkward engine design with only two air-cooled cylinders, one of them shrouded. An AFR of 14.7:1 in closed loop mode would not be such a problem except the rear air-shrouded rear cylinder often approaches searing heat ratios of 16:1. This is just too hot for an air-cooled engine.

I could ask the rhetorical question about the Motor

Company's future; what will eliminate to resolve this problem, EPA mandated Stoichiometric or the air-cooling?

The performance mechanic will also use the sensors, exhaust AFR information to modify the stock ECU dictates to approach more performance friendly fuel injection producing close to 13:1 AFR for acceleration and differing ratios for other throttle and load events that will control excessive temperatures.

Before I get into the performance aspects, I will repeat that many bikes deliver right from the Factory in a slightly rich AFR condition. The result of this is that many times but not all times a simple air breather and exhaust change will not necessitate any further ECU tuning because the now freer air flowing engine grows into the richer AFR leaning it out to perhaps where it should be. It is still very necessary to check

the AFR to see if EFI fuel modification is necessary. Do not make the assumption that everything is okay until after checking. Once again, the rider will need a mechanic/ technician/dyno tuner that he or she can trust to have the skill to determine this and the morality not to sell unnecessary parts.

## Twin Cam Ignition Modification

All mechanics inherently know that ignition curves must change with fuel modification because that is the way it has always been. One overriding consideration I have learned in recent years with working on new Harley-Davidson's is not to always take the lessons of the past and extrapolate them onto the Twin Cam. Harley-Davidson, in a valiant attempt to save the 45-degree, pushrod operated and air cooled engine we love is pulling out all the stops to prevent its demise with much technology that is subtle to the rider but has an impact on preserving this traditional engine of antiquity.

I certainly attempt to apply my previous experience to the Twin Cam because this is where much valuable knowledge emanates from but I do not make the assumption it will always apply although many times it, of course does. In Evolutions, ignition changes become very necessary because a few degrees timing change can make up to a five horsepower increase. The Twin Cam changes all of this. The effective Twin Cam anti-knock (detonation), ignition system is bang on. The thinking mechanic always advances ignition timing until heat-pressure generation causes knock, spark knock, pinging or detonation, which are all synonymous terms meaning the same thing. Then timing is retarded just enough to maintain maximum power without the engine destroying detonation.

The Twin Cam anti-knock ignition system effectively does this process of advance-then-slightly-retard ignition timing

automatically.

Although, the readily available spark-ignition timing maps accessible in both the Power Commander and the H-D Race Tuner, not to be confused with the Race Fueler, are mostly unnecessary for popular minor performancing options for the Twin Cam.

Suppose, the mechanic attempts to advance the Twin Cam ignition. The ignition will instantaneously retard the timing back to just before pinging will occur.

This is not to say, that Twin Cams do not suffer detonation because they do in attempting to control the ravages of extreme heat generated by mandated lean air-fuel ratios. I can only imagine if a Twin Cam ignition lacked the anti-knock component as to what injury would ensue.

## Electronic Control Module Modifiers

The goal of all the following modifiers is to achieve the best air fuel ratios (AFR) for both an efficiently running engine and for extra performance needs.

The fuel and sometimes-optional ignition modifiers may be input, which installs before the computer ECU or output that installs after the ECU. These add-ons might be actual hardware, software, or both. They may be manually adjustable or adjustable with download maps right off the Internet. Finally, fuel modification may involve simple tricks to fool the electronic control module (ECU), like changing the air temperature sensor for an adjustable one or simply relocate the stock one to a cooler place or activate more fuel using the stock ECU settings with larger injectors or higher-pressure regulators.

Some riders or mechanics may stack more than one modifier accidentally or hoping to achieve cumulative extra power but for the most part this does not work to increase power exponentially because once the AFR is optimal for acceleration and cruising, there is nowhere else to go. The modifier does not increase horsepower. The AFR ratio the modifier changes is what increases the power.

The engine needs the correct AFR in order to burn the air/ fuel to build pressure, driving the piston down to create the most torque. Torque is real twisting force whereas horsepower is an arbitrary mathematical function of torque multiplying rpm. If the AFR ratios are already optimal, there is no need for a fuel modifier. Generally, optimal AFR for acceleration is 13:1 and cruising is slightly below Stoichiometric 14.7:1 whereas there is most complete combustion and thus EPA dictated at 14.7:1 AFR.

However, in an emissions control world, a modifier is usually of benefit. Stacking may inadvertently happen with say a Screamin' Eagle download or burn already in the ECU computer and then adding an external modifier like a Power Commander. At Heavy Duty Cycles, we have found that sometimes a burn will complement a download map on a difficult bike-tuning project…but ya gotta know what yer doing.

## Horsepower and Torque Bike-to-Bike Consistency

Dyno operators find that Japanese and some European sport bikes have very consistent torque and horsepower output that rarely vary more than one percent from bike to bike of the same kind.

I think Harley Twin Cams have looser production tolerances. For example, cam timing is not exactly the same on every Twin Cam, possibly varying by a few degrees one-way or the other. This will give about a 10-percent variance in horsepower and torque up or down 5-percent, either way. This comment is not a criticism because the Motor Company does not produce Twin Cams for racing.

Therefore, extensive dyno testing of production Twin Cams indicate that most TC88's produce about 60 horsepower varying about 5% up or down from this average figure. Torque is usually in the mid-seventies but can vary from the low to high seventies. There is much room for improvement here in performance since one horsepower-per-cubic-inch is a reasonable quest. In fact, many of us are not content with this and demand more than one horsepower-per-cubic-inch.

Moreover, AFR ratios may bounce around a bit also, which is not at all surprising since they dictate power to a large

degree. Therefore, fuel modifiers and sometimes those with an ignition component have much benefit to most Twin Cams.

## Modifying the Electronic Control Module

There are many ways to modify the stock ECU directives to fuel delivery and sometimes ignition timing.

In general, input modifiers that interrupt and modify sensor input information to the ECU will change both fuel delivery and ignition timing because the tables for each mutually link inside the computer.

Output modifiers that locate after the ECU computer but before the EFI can only modify the fuel table and not the spark ignition although there are some output modifiers like Power Commander that have a separate ignition component

in their mapping software.

## EFI Chips, Downloads, Burns

A chip is an actual hardware device as opposed to software one that is an integrated circuit memory that plugs into the ECU circuit board replacing a stock cold emissions chip with a hot chip. Unfortunately, the H-D circuit board is inaccessible and sealed inside the ECU. Therefore, there are no chips for a Harley-Davidson like those available for upgrading car performance.

However, the Factory S.E. downloads fulfill much the same function. These are downloaded fuel maps to burn into the ECU. They may also extend the rev limiter to allow the bike to rev higher. Some may also change ignition timing. Screamin' Eagle Stage I and Stage II are limited to minor intake, engine, and/or exhaust changes.

If using TC88 Stage I burns, only do so to increase the rev limiter. This download burn will not richen the fuel AFR, in my opinion. I cheat and use the 1550cc burn in the 1450cc TC88 because this raises the rev limiter, which I don't really care about at this level of performance but will also richen the AFR, which I do care about and want. Downloads will only work properly with specified changes, which coincidentally are exhausts etc. sold by Harley-Davidson.

It is a big performance world out there and H-D upgrade-components as good as some of them are only filling a small part.

The big disadvantage of burns is that they will not work properly with ongoing and different performancing changes.

If Twin Cam production tolerances were much tighter like the sport bikes mentioned above, then a download would affect all the Harley Twin Cams equally. However, a fixed download will work wonderfully in the middle of the tolerance range provided only the specified H-D components are in use. Remember, downloads cannot adjust like other fuel modifiers as they are fixed.

What happens to the Twin Cams that are 5% under or 5% over in production tolerance? You tell me! I am not a big fan because of limitations and inflexibility.

## Piggyback EFI Modifiers

These add-ons work in conjunction with the stock electronic control module. Most do not change the H-D stock fuel maps but trick the ECU into thinking input information is

different. These controllers plug or splice into the OEM wiring harness between sensors and the ECU. Most are now plug in for ease of installation. This is also necessary, for riders to unplug them before taking their bikes in for warranty service to anal dealers that may void warranties or those dealers angry that a rider bought something from someone else other than them.

Hence, the term input controller or input fuel modifier, input modifier or simply add on. They interrupt and modify sensor information destined for the ECU. This does not alter the ECU map or tables. However it does affect which memorized cell the ECU enables to deliver fuel and linking spark ignition. This modified information fools the ECU into usually extending the time fuel-delivery-injectors are on.

For example, the input controller may take accurate sensor temperature information destined for the ECU and alter the temperature reading to trick the ECU into providing more fuel

to the engine via extending injector on-time to create a richer AFR approaching 13:1 for improved acceleration down from a leaner AFR.

## Output EFI Controllers

Many output controllers are simply pulse extenders.

The Screamin" Eagle Race Fueler, not to be confused with the SE Race Tuner (www.harley-davidson.com), Techlusion (TFI) (www.dobeckperformance.com), the Digital Fuel Optimizer (DFO) (www.customchrome.com), Vance & Hines Fuelpak (www.vanceandhines.com), and Kuryakin Wild Things FI (www.kuryakin.com), controllers are good examples of pulse extenders. Techlusion and DFO are much the same with different names as one is the private branding of the other. This is not a big deal since they do the job. These units

richen the OEM stock fuel curve.

However, input controllers also have the ignition component because the stock ECU fuel and ignition tables link with each other.

Potentiometers

Many have pots (potentiometers) that are adjustable by screwdriver or thumbnail. A number of the potentiometers have new generation models converting to digital. The digital upgrades from the pot methodology replace with buttons, a mode select and plus or minus functions. The pots or buttons adjust rpm and fuel mixture. However, they cannot change the stock ECU rev limiter setting since they are output units and locate after the ECU.

They adjust rpm by, which pot or digital button controls,

which rpm range. Specifically, and more correctly they adjust the rpm transition points between the rpm band pot-controls that is if there is more than one potentiometer. Each pot is similar to carburetor jetting for low, mid and high range, which is why these units are so popular because there is a familiarity with the adjustment process.

EFI fuel injectors squirt gas towards the intake valve or at least they should. Early H-D Magneti-Marelli (MM) EFI, for what reason, I will never understand does not squirt fuel towards the intake valves. Of course, I do not understand their air leaks and overly small induction throats either.

I fully understand why Evo and Twin Cams outfitted with Magneti-Marelli EFI from 1995 to circa 2001 are dogs in the performance department. The successor Delphi EFI is wonderful compared to MM and improves year-by-year.

## Pulse Width

EFI injector squirt volume is consistent. Pulse width is the term for the length of time the injector stays open or is squirting. Therefore, a pulse extender increases pulse width to squirt extra gas by expanding the time the injector stays on.

These fuel modifiers have a common efficiency that makes them worth using. The pot(s) or digital buttons are linear or load-based. This is the bonus since linear-based ensures that less fuel adds at lower rpm levels than higher ones where more fuel modifies the lean AFR to a richer one. Lower rpm levels have more fuel and thus a richer AFR. If extra fuel adding, lowers the low-to-mid-range AFR too much then torque numbers will suffer.

## Choosing the rpm Transition Point

Another good feature is that these modifiers allow the owner or tuner to choose the transition rpm point. This can be critical to performance because adjusting the AFR above and below this rpm point is a separate function. There is one transition point between two pots allowing for independent adjusting in two ranges, one above and one below. The Race Fueler has two transition point choices between three potentiometers allowing much more flexibility with independent AFR tuning in three chosen ranges above and below the two points. Four pot/digital button modifiers will have three transition points allowing the most flexibility with independent AFR tuning in four chosen ranges above and below the three points.

Extending Torque Curves

Extending torque curves is one obvious benefit because of choosing the transition point near its current end. I will be

able to richen above the transition point, thus extending the curve but keep the current AFR richness below the point without decreasing torque by adding too much fuel in this area thus lowering the AFR too much.

I wanted also to demonstrate that these supposedly foolproof pot modifiers actually require careful thought and strategic thinking to use them to their fullest potential. The Vance & Hines Fuelpak can also decrease fuel, which is handy when doing cam work. The others mentioned here can only increase fuel delivery.

## Will these Modifiers Work Well with Cam Changes?

These pot/digital button output modifiers work very well with air breather and exhaust changes. However, cam work can complicate air-fuel curve needs.

I feel that the above units including TFI, DFO, Wild Things FI and the H-D Race Fueler are not good choices for Harley's with cam work because they cannot divide the rpm power band into small enough ranges to accommodate the new air-fuel curves without more transition points.

When progressing beyond air breather and exhaust changes, I favor the use of map-style devices with engine management software like Power Commander, Screamin' Eagle Race Tuner or Direct Link or self-tuning modifiers like S&S Cycle, Inc.'s carburetor IST, which is the acronym for Intelligent Spark Technology or Terry Components Terminal Velocity.

## Engine Management Software

This is sophisticated stuff, whereby actual changes to ECU fuel and ignition tables create new maps in addition to

altering rev limits. Power Commander is the long time king of this methodology with the newer Screamin' Eagle Race Tuner, which also gives control over the ignition and spark knock sensors. Another example is TechoResearch's Direct Link (www.technoresearch.com).

## Power Commander

Many think the Power Commander is only an excellent fuel map modifier since it is an output system after the ECU and therefore cannot tie in the ignition curves with the fuel map. The Power Commander has a separate ignition function. Input modifiers take advantage of the fact that the ECU ignition tables link directly to the fuel mapping.

Thus ignition changes coordinate with the fuel map modification. This is not an artificial change overlaying the existing ECU fuel table or map but a totally new one suitable

for performancing changes to the motorcycle.

There are thousands of downloadable maps for the mechanic to install. It also gives mechanics the ability to modify an existing download map to say, add fuel to a flat spot in the rpm/engine load range plus the capability to create a new map altogether. Many a rider and mechanic alike have trouble working with these software packages but this is where tuning is currently at. Once, surpassing the intimidation factor and the mind-set adopts this type of technology, it actually becomes quite simple.

The Power Commander (www.powercommander.com), from Dynojet Research Inc. (www.dynojet.com), is a map-base device, which is for all intents and purposes an add-on computer that connects in series after the ECU. The Power Commander tunes via another computer, usually a laptop, or

even a palm pilot. The fuel mixture AFR adjusts for various engine speeds with gradations every 250-rpm and throttle positions from 0% to 100% via a specific downloadable fuel map matching changes made to the motorcycle.

Although normal for most Harley's, annoying exhaust popping during deceleration, can in almost all cases correct easily via Power Commander tuning.

If dyno tuning with a Power Commander, the appropriate map consisting of two fuel tables and two linked ignition tables downloads in less than five seconds while the bike is running. When purchasing a Power Commander, a CD is part of the package that contains the most popular maps for current parts popularity and use in many combinations.

The total bank of ever increasing maps is available for free

downloading from the Power Commander Internet site. As new maps continuously develop, they are added to the Website.

Power Commander also allows the tuner to view the last map installation. This is valuable since it may only need some tweaking to make it perfect for the motorcycle and customer needs. The Power Commander, plugs in-line with the stock ECU and EFI in the Harley wiring harnesses using stock connectors so splicing is avoided.

It allows a full range of fuel adjustment + or – 100% over stock. Adjustment accomplishes with Dynojet tuning, Windows based computer or by adjusting the Power Commander PCIII USB face plate buttons. The faceplate button adjustment has a start-up delay feature. The owner/ tuner has time to start the bike before turning the switch into button adjust mode. Once in this mode, the ranges of the adjuster buttons display in the software by connecting the

Power Commander via its USB port to a computer. Simply press one of the buttons and the computer screen will show the button ranges.

The beauty for those concerned with warranty situations is that the Power Commander unit removes easily by unplugging its connectors from the H-D wiring harness and re-plugging the stock connectors.

There is also an expansion port in the PCIII USB unit that allows easy connection for upgrades or accessories; such as the newly available ignition module, quick shifter for the race fans, a multifunction hub, a pressure sensor display and LCD display unit, and automatic map retrieval and display of the currently installed PCIII USB map.

There is a 9V power up adaptor, which allows the owner/

tuner to program and interface with the PCIII USB on their laptop or desktop without connecting to the bike.

With the accelerator pump, which, of course is not a traditional carburetor accelerator pump but an electronic one, the owner/tuner can adjust the amount of fuel to add, the number of engine revolutions during which the fuel increases and the sensitivity of throttle movement that activates the accelerator pump.

Any Power Commander tuning is not permanent so as not to impact stock settings when unplugged. The Harley will revert back to the stock Factory settings. An individual cylinder adjustment feature, the cylinder trim adjustment allows the tuner to offset fuel to each cylinder, an especially nifty procedure for the hotter running rear cylinder, which can now cool with a little fuel enrichment. In fact this individual cylinder mapping now occurs throughout the whole map's rpm/throttle position range.

Harley-Davidson PCIII USB features now include an ability to increase the rpm limit over stock without using a Harley-Davidson flash, burn or download called Rev

A second recent feature, Start Fuel, allows an extra adjustable fuel amount on EFI startup to ease starting particularly useful with big-inch engines without making the idle too rich once running.

The Power Commander supplies with a base map in the microprocessor that will provide an improvement for the majority of combinations matching the map description, i.e. stock bike or performance exhaust and air filter. Fine-tuning can go from here with other map downloads or custom tailoring a map. Even a stock map will invariably give performance increases because the stock ECM put emissions

limitations ahead of performance ones.

## Screamin’ Eagle Race Tuner and Direct Link

The Power Commander is a map-based device while Direct Link and the Screamin’ Eagle Race Tuner (SERT) are programmers from a map base. A programmer allows the tuner to go into the stock ECU and change some of the software with new mapping. This is a recent innovation as this type of programming was not available in the aftermarket before. Again, unlike other map-based units like the PCIII USB, once the software data transfer or download accomplishes with a programmer, there is nothing to indicate anything has changed. In other words, there is no unit installation before or after the stock ECU.

The Direct Link and SERT software allows the tuner to

modify the EFI injector pulse-width, spark timing, and other calibration values into the flash memory of the ECU like starting, warm-up fuel and rpm functions. A cool feature is to set the injector size that the ECU thinks it is working with. This, of course, is a way to enrich or lean all the fuel cells of the map at the same time. Rev limits are adjustable to below stock up to wherever the tuner

Direct Link also has an optional wideband controller for each cylinder that will obviously maximize tuning results and thus performance.

Both work from a zero base map, the same as Power Commander. However, I feel they both become cumbersome, SERT more so than Direct Link, relative to Power Commander.

SERT has separate maps and data monitoring mode programs that cannot share an interface with the motorcycle. I

think SERT is awkward by requiring alternate stopping and restarting one to use the other back and forth. I believe Direct Link is not quite as unwieldy in navigating its menus and functions are easier to navigate than SERT but still not as easy as Power Commander. SERT's air-fuel adjustment range in basic mode limits to about one-fifth of total while Direct Link does not impose this seemingly arbitrary limitation. Both SERT and Direct Link have a download time of about 30-seconds with the ignition switch on and the engine not fired up followed by turning the ignition switch off for 10-seconds, which initially seems fast.

However, with dyno tuning this means that the running bike must switch into neutral, turn it off, perform the adjustment, turn the bike on, start it up and retest for the next adjustment, which is a laborious process compared to Power Commander dyno tuning. Starting with the previous map is also more difficult since SERT and Direct Link are write-only

programs.

Both their calibrations are re-flashing maps via serial programming into the ECU.

However, modifying front and rear cylinder fuel & spark tables allows sophistication for the tuner especially when the H-D rear cylinder shrouds from airflow and runs hotter than the front.

SERT has an advantage for the consumer since the different Stage I and Stage II download burns including for rpm limiter settings includes with the purchase. If download maps match and tune your bike then everything is great.

The disadvantage is that if further dialing in is necessary it can be a hard slog.

Power Commander is transferable from one bike to another as it is a simple disconnect and reconnect.

SERT requires an interface module hardware key. Insist your mechanic give it to you along with the most recent modified download on a disc for the next tuner. No key equates to an inability to retune and the recent download is necessary to save much time.

Regardless of criticism, all of these sophisticated maps basing software and programming modifiers are state of the art relative to only a few years ago. They allow the tuner latitude previously unavailable. They will correct wavy AFR lines into a straighter desirable plane that potentiometer piggyback controllers cannot. These units will resolve AFR problems that other less sophisticated modifiers cannot begin to broach.

It is a long way from the carburetor tuner with screwdriver in hand.

## EFI Modifier Technology Changes Fast

Anything I write today about fuel or spark modification may supersede by newer types of modifiers coming into the marketplace. However the principles are the same.

What is available in this area of EFI tuning changes quickly for two reasons;

- One, increasing emissions controls dictates creates new industries and technologies to meet them and to tune for them.

- Secondly, much technology exists on the automotive side not yet available to the Harley rider. In other words there are innovative manufacturers transferring existing technology into Harley user-friendly modes continuously.

RideMaxx EFI Controller

RideMaxx, from American Micro-Fuel Device Corp. (www.americanmicrofuel.com), is a new EFI controller in the marketplace for Harley. It features a wireless remote control module that communicates with a handlebar module. Like Power Commander, it plugs into the Harley ECU but it supplies with pre-loaded power and fuel consumption maps.

RideMaxx claims up to 15% increases in horsepower in power mode. This unit, as all of them do, releases horsepower suppressed by emissions controls that place best combustion

possible in priority above best performance.

The RideMaxx also has a nifty conservation package that conserves fuel on long distance rides where it counts. The economy mode claims fuel conservation of up to 25-miles per tank.

A third mode, which is the automatic one, alternates between the two settings of conservation and power. This provides a live and unspoiled performance improvement for normal riding. Of course, when the race is imminent, power mode is the choice.

The battery-powered wireless remote handlebar module has two buttons, power and economy. This allows the rider easy and safe access to change maps while riding. RideMaxx equips with Bluetooth enabled technologies. It offers wireless tuning, data logging, and the ability to display bike

information on Bluetooth enabled devices such as handheld PDA's and computers. RideMaxx use of technology allows for wireless downloads of maps without any access to the control unit itself, as well as wireless access to fuel tables, and tuning on the fly.

The current downside, that promises correction shortly as of this writing is that

RideMaxx is only currently available for H-D open loop EFI equipped Touring models and Softail models. Therefore RideMaxx is currently available for EFI equipped Harley's up to 2006 excepting the closed loop Dyna model EFI. Closed loop attempts to constantly reach Stoichiometric ratio of 14.7:1, the ideal ratio for reducing emissions by most complete combustion. Closed loop uses are for cruising and fast idle, where fuel conservation is possible. An open loop system is not emissions friendly but is necessary for idling (rich condition) and acceleration because power requires

more fuel relative to air (12.8:1 to 13.2:1). Modern EFI employs closed loop for fast idle and cruising and reverts to open loop for cold idle and acceleration. Of course, as the reader knows, closed loop is the EFI system in use for 2006 Dyna models and all 2007 and later Twin Cams and Sportsters.

RideMaxx indicates that units for closed loop Touring, Softail, Dyna, Sportster, and V-Rod models, models will be available soon, surely by the time this book publishes.

## I Apologize for those EFI Modifiers I Left Out

I apologize for not covering every fuel and fuel/spark modifier currently on the market but that would be a tome in itself. I try to provide enough information that the reader can research other modifiers and scrutinize their strengths and weaknesses effectively with the analytical knowledge gain

from this book.

## Replacement ECU's

The sophisticated Daytona Twin Tec (www.daytona-twintec.com), ignition modules come with their own Engine Management System (EMS). They work extremely well but are sophisticated for ordinary installers and expensive because of leading edge technology for Harley-Davidson and more comes with them.

## Terminal Velocity

Terminal Velocity is an input system as is Race Tuner, where the signal from the oxygen sensor modifies the stock ECU by sending information to modify fuel. The fuel table in the ECU directly relates to the ignition table so both fuel and

spark is modified.

Output fuel modifiers on the other side of the ECU can only modify fuel by increasing pulse width of the injector although the sophisticated output Power Commander has an ignition component.

Terminal Velocity plugs into the stock wiring harness although early models inconveniently spliced-in. It utilizes stock O2 sensors on later model 2006 Dyna and all 2007 and later Twin Cams. T.V. supplies an O2 sensor(s) and weld in sensor bung(s) for earlier model EFI Harley's. The bung is easy to locate out of sight for the fashion conscious.

As I have relayed before, the Terminal Velocity is a self-tuning unit that requires nothing of the mechanic or rider after installation. I can feel changes immediately after initial riding but full tuning can take as long as 50-miles riding.

Ongoing performance changes, additions or deletions such as changing exhaust pipes, breathers and cams require zero tuning as the Terminal Velocity constantly and automatically tunes.

## Changing EFI Fuel Delivery without a Modifier

There are other ways of fooling with the stock ECU without a controller/modifier such as replacing the engine temperature sensor with a variable resister; relocating the temperature sensor to a cooler place or installing larger EFI injectors and/or increasing the fuel pump pressure.

## Altering the Air Temperature Sensor

Changing a temperature sensor with a variable resistor allows the operator to set an artificial engine temperature. This has a huge effect on fuel delivery. The resistor sends the new temperature (lies), to the ECU in order to increase or decrease fuel. Of course, in performancing an emissions motorcycle we want to increase fuel relative to air to bring AFR down into the 13:1 range.

The problem is that, whoever sets the resistor temperature must have extensive engine tuning knowledge because if the setting is inaccurate, too much fuel or a rich AFR will cause a decrease in torque and horsepower particularly in midrange and a too lean AFR may also affect performance negatively but also damage the engine. The fuel mixture will increase right across the map. This may pose a difficulty because it is in high rpm areas that need richening more than midrange. In fact, midrange may work perfectly but may not if the AFR richens further. Furthermore, temperature gauges will be off

and not read accurately.

Relocating the temperature sensor to a cooler place like inside the air breather will give a more predictable result for those of us who are not rocket scientists. Engine temperature is a powerful tool that will ensure the ECU changes fuel delivery.

Increasing the fuel pressure regulator to increase fuel delivery to the engine will not have as much affect as a variable temperature resistor. Potentially, increasing pressure will result in about a 6% or less fuel increase.

Bigger Injectors

This change will also alter the fuel increase across the whole map so while high rpm AFR will improve; it may be at the expense of a torque robbing overly rich AFR in the

midrange. Injectors measure in pounds-of-fuel-per-hour. Increasing their size increases fuel delivery accordingly. This is an effective way of simply dealing with typical closed loop EFI high rpm lean AFR since the ECU will pump much more fuel in this area than midrange. However, if the injectors are too big, then mid range will overly richen and lower torque.

EFI Injector Flow Maximum Engine HP Potential

Example; injector flow rate 25 lbs/hr; H-D stock EFI (2 injectors)

Injector flow rating x 2 x injector number

25 x 2 x 2 = 100 HP potential

A narrowband sensor is essentially a switch that flips on and off at 14.7:1 AFR. A wideband sensor is the same as those in use on dynamometers to measure AFR; they measure from 10:1 AFR to 18:1 giving a linear output signal. I should re-mention that the richest AFR requirement for acceleration and cold idle is about 12.8:1 going up to about 13.2:1 and averaging 13.0:1 for good acceleration. The leanest condition that may help cruising and hot idle conditions is 15.0:1 down to a richer but still lean 14.5:1. Therefore necessary capability is from about 12.8:1 to 15:0:1 on a Harley-Davidson.

I suspect that some modifiers shut out the narrow band sensor to preset an AFR with their own mapping or map basing system. In these cases of EPA-illegal tampering O2 sensor simulators are necessary. The tuning device also needs to keep the Check Engine Light or Service Soon Light from turning on.

There is nothing wrong with engine function in these situations but the light is distracting, especially for the rider that does not understand why it is on. Of course, if a real problem develops, the light cannot warn the rider since it is already on because of the fuel modifier.

O2 sensor simulators ensure that the Twin Cam ECU does not make any changes due to O2 sensor voltage input that might affect adjustments to the fuel injectors and ignition advance/retard timing signals outputting from the ECU into the fuel modifier that wants to alter them for performance reasons. Voltage or more correctly millivolt pulses from the O2 sensor give exhaust composition readings to the ECU. I short; the O2 sensor simulators prevent the ECU from receiving true voltage readings and allow simulated or false readings to trick the ECU.

However, narrowband sensors can still work well with performance fuel or fuel/ignition modifiers because it will seek a mostly acceptable 14.7:1 AFR for cruise and hot idle conditions and shut off for acceleration and cold idle starting. Narrow Band oxygen sensors cover the whole range of performance options because change does not happen really fast. The demonstrable proof is that many automotive performance modifiers have been working well for a few decades with narrow band sensors.

I have stated many times that wideband is very necessary with forced induction, (supercharging, turbocharging, nitrous oxide), where boost can lean out an engine very fast; sometimes too fast for a narrow band sensor to react before super-lean AFR's can occur. However, some cars with superchargers and turbochargers can and still do use narrow band sensors because boost is low from between 6 to 10 lbs. Higher boosts need the safety of a wideband sensor. Wideband

sensors can react in a timely fashion to rapidly climbing lean AFR's.

Narrow band will work fine with most normally aspirated engines.

## Conclusion

Hopefully, this extended discussion will give the reader enough information to make educated decisions if fulfilling their wants; whether it is simply for an efficient running engine or for the go-fast crowd.

Many think they understand carburetion, which in my opinion is more complicated than electronic fuel injection.

Surely, EFI becomes complicating with the necessary knowledge for fuel and spark modification, but then, so does carburetion with larger manifolds, venturis and much jetting and needle adjustments. Then, of course, carburetion requires extended knowledge of ignition advance and retard timing curves and their adjusting also.

Let us not forget the simple things in mechanics that many gloss over in resolving the complicated. The air cleaner section demonstrates that the simple may complicate very quickly. What is the point of expensive carburetor or EFI upgrading if air cannot flow into them efficiently with high laminar volume?

Lastly, I am annoyingly repetitive on some points purposefully as I want to drive basics home that the reader can easily build on.

1 Tim Doherty, Doherty Machine. Email interview July 16, 2007.

2 Ann Christoffer, Test Technician at S&S Cycle, Inc. Email interview July 25, 2007.

3 Report of the House of Representatives. United States of America. House Report No. 93-1197, Congress Session.

4 Tim Doherty, Doherty Machine. Email interview July 16, 2007.

5 Doherty Machine. Website, http:/ (www.dohertymachine.com). Magazine articles.

6 Chris Maida. American Iron May 2003. Issue 169. Pages 46,

48.

7 It's The Real Deal, Doherty Power PACC. Easyriders August 2002. Issue 350. Pages 130-132

8 Tim Doherty, Doherty Machine. Email interview July 16, 2007.

9 Scott Sjovall, VP of Product Development for S&S Cycle, Inc. Email interview July 9, 2007.

10 S&S Cycle, Inc. 2007. '07 Performance Guide for 1999-'06 Big Twins Twin Cam 88® and Twin Cam 88B™ Engines. Page 43.

11 Scott Sjovall, VP of Product Development for S&S Cycle, Inc. Email interview July 9, 2007.

12 Ann Christoffer, Test Technician at S&S Cycle, Inc. Email interview July 25, 2007. 13 Lee Wimmer. Wimmer Custom Cycle. Email interview July 12, 2007.

14 Ibid. July 12, 2007.

15 Wood Performance Carburetors. Website, http:/ (www.woodcarbs.com)/air.htm, air filters.

16 Scott Sjovall, VP of Product Development for S&S Cycle, Inc. Email interview July 9, 2007.

17 S&S Cycle, Inc. 2007. '07 Performance Guide for 1999-'06 Big Twins Twin Cam 88® and Twin Cam 88B™ Engines. Page 34.

# Chapter X

## The Power Train

The powertrain is a descriptive term given to a series of attaching components that transfers torque back from the engine through the shock absorbing compensator sprocket to the clutch sprocket via the attaching primary chain, into the transmission via the clutch to the transmission pulley and back to the rear pulley via a belt drive. The pulley attaches to the rear wheel. Therefore, the torque transfers to the contact patch of the rear tire meeting the asphalt.

Gearing ratios, first in the primary transfer system, modifying by different gears in the transmission, the

secondary gear ratio, which is the relationship between the transmission pulley size and the rear pulley size and lastly the final drive ratio that determines how many times the rear tire turns a complete revolution relative to engine rpm.

1. The 2007-2008 TC96 engine rotates 2.79 times for every sole revolution of the rear wheel.

2. The 1999-2006 TC88 engines, except the 2006 Dyna models, rotate 3.15 times for every single turn of the rear wheel.

3. The 2006 Dyna and the 2009 Twin Cam engine rotate 2.88 times for every rear wheel revolution.

The 2006 Dyna and the 2009 Twin Cam has the same number of engine, clutch, and transmission sprocket/pulley

teeth as the 2007 to present Twin Cams. However, the rear wheel pulley has two more nubs or teeth than the 2007-2008 TC96 models, thus giving a different and higher final drive ratio. USA 2007-2008 TC96 Twin Cams use a 66-tooth rear pulley while the 2006 Dynas and 2009 Twin Cams use a 68-tooth pulley. Earlier TC88 Twin Cams use a 70-tooth rear pulley while other models may have differing teeth/nubs. Harley-Davidson International (HDI), with the exception of Canada that usually adheres to USA gearing bikes use wildly different smaller rear pulley sizing because of higher speed limit highways.

- Increasing off-the-line acceleration involves lowering the engine sprocket, increasing the clutch sprocket size, lowering the transmission pulley/sprocket and raising the rear pulley size.

- Increasing highway comfort and speed involves increasing the engine sprocket, decreasing the clutch sprocket

size, increasing the transmission pulley/sprocket and decreasing the rear pulley size.

Usually, only the easy labor saving sprocket/pulleys change to save expense with the rear pulley the most accommodating followed by the engine sprocket and then the transmission pulley/sprocket. Of course, if the primary is off for other reasons, the transmission pulley becomes easily accessible. I will explore gear ratios more thoroughly as I proceed and in a following section, Gear Ratios.

A Harley-Davidson, like most vehicles and a lot of machinery requires a mechanism to interrupt this power flow to the rear wheel for stopping and starting without stalling the engine or damaging transmission gears.

This apparatus is a clutch. The clutch is an uncomplicated

way to interrupt power transfer without damaging parts or stalling the engine for the purpose of stopping and smooth acceleration when proceeding forward again.

## The Engine Compensating Sprocket

Big Twins, 1955 to present use two basic types of engine-to-transmission power shock absorption devices. The compensator assembly cushions the pulsing energy of the motor to protect the powertrain going back through the transmission.

Sportsters began compensator use in 1957 to the mid-seventies but then began alternating their application with direct drive engine sprockets.

The downside to these various absorption mechanisms is that they suffer some power transfer loss from the engine to

the rear wheel where the tire patch contacts the asphalt. Many times when performancing Big Twins and subsequent Twin Cams up to 2006, I will change the engine sprocket to a larger or smaller one to move the power band up or down. I will also eliminate the compensator assembly for a direct drive sprocket to send an extra bit of torque back to the rear wheel.

In general, I use a good compromise 24-tooth engine sprocket on these years but will go down to a 23-tooth sprocket for faster acceleration from a stop or a 25-tooth sprocket for more pleasant highway cruising. A 26-tooth direct drive sprocket faces space limitations. It also gears the bike too high for my liking. Conversely, there is room for decreasing sprockets but reducing to 22-tooth or even 21-tooth engine sprockets is for serious city riding or should I say hotrodding.

The 2006 Dyna and all 2007 to present Twin Cams are in a different ballgame with the Factory supercharging the gear ratios with a massive 34-tooth engine sprocket combining with a big 46-tooth clutch sprocket.

Current performance re-gearing of these new Twin Cams involves mostly the transmission pulley going a few nubs/ teeth up, down, or to a lesser extent changing the size of the rear pulley, which has 66-nubs or teeth for 2007-2008 and 68-nubs for 2009.

Big Twins, 1994-2006 except the 2006 Dynas uses a formerly big 25-tooth engine sprocket combining with a 36-tooth clutch sprocket. Pre-1994 Big Twins use a 24-tooth engine sprocket with a 37-tooth clutch sprocket.

There are two main compensating assemblies used in the

Harley-Davidson. Only one, which incorporates a very heavy coiled spring, looks like an absorption device. The second type with variations throughout the years from 1970 to present in Big Twins eliminates the coil spring in favor of diaphragm springs.

The 1955-1969 Big Twin and all Sportster compensators use the coil-spring type device. Two connecting cams ride up on each other, transferring the power surge and compressing the attached spring to absorb the initial shock while the main thrust goes back along the primary chain through the clutch and the transmission to the rear wheel.

The cams are hollow heavy steel. Each has valleys interspersing with and flowing into crests much like waves. The high areas of the outer cam fit into the low areas of the mating cam when at rest. The spring forcefully holds them in

this position. The whole assembly holds firmly by a large compensator nut that screws onto the threaded end of the engine sprocket shaft that protrudes into the inside of the primary system.

An engine pulse will suddenly transfer torque through the compensator. This shock will cause the compensator cams will ride up on each other, compressing the heavy hold-down spring. After engine shock absorption subsides, the decompressing spring forces the cams to ride back down into there resting position. This shock-absorption process happens faster than the human eye can follow thousands-of-times-per-minute. The remaining types of compensators are similar in function since they still use the interacting rotating cams.

Knocking Cams

Two noises, one very loud and sinister-sounding, will

ensue with worn compensator parts that in effect create unwanted space between them. Rattling absorption cams sound like the engine is coming apart.

Another lesser noise may occur and that is a rattling alternator rotor. The alternator comprises of a stator and rotor. They sandwich between the sprocket shaft extension that the rest of the compensator parts work from and the outer motor case on the inside of the primary.

Up until the 2006 Dyna models, the rotor's splined inner fits on the corresponding sprocket shaft spline and rotates with the flywheels and the compensator. If excess play occurs, it can cause a knock of its own. Immediate repair is necessary because the stator can consequently damage and cause the charging system to malfunction.

If on the road and diagnosing a compensatory noise, which is most easily heard when taking off in first gear, it is safe to proceed home or to a repair shop at a steady speed if close by. However, if requiring temporary repairs, it is possible to add a diaphragm spring to the compensator package to beef it up. Of course, who carries an extra diaphragm spring and the necessary tools? An easy roadside fix is to install a washer under the compensator nut to take up excess play and increase spring tension.

Another method is to get rid of the compensating unit altogether and go direct drive. Simply use an aftermarket splined motor sprocket or an original Harley sprocket from early models (40211-55).

Compensator-Alternator Rotor

There are other big changes to the new 34-tooth compensator-engine sprocket parts on the 2006 Dyna models and the 2007 to present TC96 Twin Cams. A compensator that has always been a part of the engine sprocket originally included a big coil spring on the pre 1970 Big Twin absorbs energy shock from the engine back through the powertrain. Since 1970, the spring portion of the compensator consists of a series of fixed and thick diaphragm spring plates. The new 2006 Dyna and 2007 Big Twin 6-speed models have a 4-spring pack for both domestic and H-D International models (HDI).

The compensator portion was a component assembly unto itself that fit between other mating parts in front of the alternator adjacent to the outside of the engine casing inside the primary housing. Now the rotor portion of the alternator combines with the compensator as one part that is finely multi-splined (24-splines), compared with predecessor models (10-splines, ⅛”-wide), in the middle to fit over mating 24-fine

splines on the engine output sprocket shaft.

First introduced in the 2006 Dyna; the rotor capacity increases for 2007. The welded together compensator-rotor unit rotates with the engine with the rotor spinning over the fixed stator, the other half of an alternator to produce current to charge the battery. A new 40-amp alternator assembly upgrades the Softail and Dyna models. The magnets increase to twelve on the rotor. This stator and rotor assembly will retrofit to the 2006 Dyna models only.

In order to service as in replace the rotor, stator or both on the rubbermount FLHT; removal of the inner primary was not necessary; it is now.

Compensating Sporockets

Big Twins from 1955 to present and Sportsters from 1957 to

the mid-70s use two basic types of engine-to-transmission power-shock-absorption-devices. The absorption device cushions the pulsing energy of the motor to protect the powertrain back through the transmission. The downside to these various mechanisms is that they suffer some power transfer loss from the engine to the rear wheel. This is another small reason that the Sportster is faster than the Big Twin since many models are direct drive with only a sprocket and not the added compensating sprocket assembly. Pictured on top is an early Sportster coil-spring compensating sprocket assembly. The Big Twin coil-spring compensator is very similar. However, the Sportster uses a triplex primary chain and thus has three rows of sprocket teeth while the Big Twin uses two rows with bigger teeth and chain. When compensating sprockets wear, they begin to make an insidious noise. As the wear becomes worse, the noise will sound like the engine's bottom end is about to grenade. All Sportster and Big Twins from 1955 to 1969 compensators incorporate a coiled spring (bottom left, top picture), as an absorption

device. Two interlocking cams ride up on each other, transferring the power surge and compressing the attached spring to absorb the initial shock while the main thrust goes back along the primary chain and back through the powertrain. After the shock, the cams will rotate back to their resting position as the spring decompresses. The cams are part of the sprocket bottom right and it's mating part bottom center, top picture. Bottom picture is one of a series of 1970 to present compensating sprockets using diaphragm springs pictured in their housing, bottom-picture-center-left. To the uninitiated, these springs look like flat pieces of steel and it is difficult to see how they would compress let alone absorb shock, but they do. The center row shows the two mating cams that ride up on each other. The top row shows the center guide that slides onto the splined sprocket shaft along with the big hold down nut. The bottom sprocket is a direct drive in use up to 1955 on Big Twins. Performance mechanics will use this sprocket to replace the compensating sprocket assembly to gain a little extra power transfer back to the rear

wheel.

Screamin’ Eagle Compensator Fix: 2008

This beefy compensating assembly, part number 40274-08, works so much better than the stock original TC96 compensating sprocket assembly 40296-06 and the improved one, 40296-06A. If I may go so far as to say, in comparative terms, it showcases the weakness of the stock early and upgraded late TC96/CVO 110 compensators I do not think I would be exaggerating.

Obviously a new alternator rotor comes with the upgrade since it is integral with the compensator housing and springs.

- The original six-speed models have a four-spring pack compensator-rotor for both domestic and H-D International

models.

- The S.E. upgrade incorporates two very large outside spring washers with one set about the diameter of unit, a medium size spring washer and one smaller one. The design appears to be progressive in order to complement larger motors down to the relatively smaller TC96 units. The compensator ramps are also changed.

Start up bang should be a thing of the past for many and along with IDS should make bike even better.

Compounding the problem is that apparently some starter gear rings located on the clutch hub outer casement were not heat-treated correctly. This of course, will cause ring gear tooth damage and teeth loss. This situation is now corrected.

This new S.E. compensator is massive and in my opinion should been used in the first place on the TC96 models and especially the CVO 110. However, it is standard equipment on the 2009 103c.i FLHTCUTG Tri Glide Trike

The good news is that it will retrofit to,

1. 2006 to present Dynas,

2. 2007 to present Touring models,

3. Most Softail models except the FXCW, FXCWE and the FXSTSSE3.

This S.E. compensator will be the best yet at controlling the pulses of a wide variety of build sizes. Some owners report

their bikes feel stronger, with no startup bang. Others that have installed the S.E. compensator on 2007 models without the IDS rear pulley dampener system report a slight tick sound. This would suggest that the new compensator is successful in controlling engine pulses traveling down the powertrain and reverberating back up into the transmission thus reducing fifth-gear click substantially. To take this a little further, a reinforced and stronger compensator may have been the cure all along making the IDS largely unnecessary. However, being a typical Harley rider where more is always better, it is nice to have both.

Inner Primary

Removal of the inner primary is now necessary on all Big Twins in 2007-present to service the rotor and stator. A new primary chain (#40037-07 FLHT, #40036-06 FXST/Dyna)), connects the engine sprocket to the clutch shell sprocket, which moves to the outboard of the housing from its

traditional inboard placement since 1965.

## Is the TC96 Gearing Too High?

I will reiterate that the marriage of transmission gear ratios with the primary and secondary ratios of the TC96 is the best ever.

Nevertheless, I feel there is a problem with newer riders lugging these bikes because of the improved ratios and the higher than normal speeds they require. I, as an experienced rider find the new ratios enhance my riding fun but exacerbate my problematic innate speeding habits. My Streetglide does not want to stay within legal speed limits. In fact, I do not feel there is a highway in North America that allows running a TC96 at a good clip in sixth gear.

However, the gear ratios are perfect for a European ride

where restrained speed is not so much of an issue.

Where I ride to work in Toronto's east end, the local speed limit is 60kph (38mph). On my previous Harley's, I would cruise along at 80kph (50mph). My pre-2007 bikes felt comfortable at both the forty or fifty mile per hour speeds. The comfort level of my 2007 Streetglide continually edges up to 100kph (60mph).

I constantly have to pay attention to reduce my speeds as the bike seeks its sweet spot, which unfortunately contravenes local laws. This is confusing to new riders. They want to obey speed limits but the bikes crave to go faster. Many wind up lugging their engines by riding at too slow-a-speed rpm for a particular gear. This of course, can easily damage the engine.

Obeying a 90kph (55 mph), or a 100kph (60 mph), speed

limit on the highway is annoying with the Cruise Drive gear ratios. There is no need for a sixth gear in this situation. As a result, lugging in sixth gear is common.

One-hundred-and-twenty kilometers an hour (70mph), was a sweet spot for my 2006 and earlier bikes on nearby 100kph (60 mph), highways. The police in my jurisdiction usually tolerate this but not more. One-hundred-and-thirty-five kilometers per hour (80mph), seems to be the new norm for my particular riding style in sixth gear on my 2007 FLHXI. The police do not tolerate this speed. Again, I must consciously monitor my pace.

This is annoying but when I start to daydream, the ride is pleasurable…until I come back to reality.

A BAKER Drivetrain Lower Gear Remedy

BAKER Drivetrain (www.bakerdrivetrain.com) has once again come to the rescue. Their cure will allow increased off-the-line response with more torque. As well, increased torque will affect performance positively throughout the rpm range.

Furthermore, use of sixth gear at lower speeds without lugging is a big benefit of the two new smaller transmission pulleys developed by BAKER Drivetrain.

The stock 2006 Dyna and all 2007 to present Twin Cams use a 32-tooth transmission pulley (40659-06), the same size as predecessor Twin Cams and Evos going back to 1994 (40250-94C). However, the pulleys are not interchangeable. BAKER claims rear-wheel-torque increases seven percent with their 30-tooth pulley and by three percent with their 31-tooth pulley when used with Harley-Davidson's Cruise Drive 6-speed transmission. Installation requires a BAKER 2¼" special pulley nut socket (94660-07BD). The smaller thirty and 31-

tooth pulleys also need a belt with one-less tooth than stock.

## Primary Gear Reduction Ratio

This ratio calculates by dividing the number of teeth on the clutch hub sprocket by the number of teeth on the engine sprocket.

$$\text{Primary gear ratio} = \frac{\text{Cutch sprocket teeth}}{\text{Engine sprocket teeth}}$$

## Historical Decreasing Primary Gear Ratios

## Sprocket/pulley Sizes

The 1000cc 1973 Sportster had a 34-tooth engine sprocket combining with a 59-tooth clutch hub sprocket giving a primary ratio of 1.74 engine rotations for one rotation of the clutch hub and the transmission mainshaft.

Early Shovelheads usually had a 23-tooth engine sprocket combining with a 37-tooth clutch hub sprocket giving a primary ratio of 1.54 engine rotations for one rotation of the clutch hub and the transmission mainshaft.

Evolution Big Twins from 1985 to 1999 usually had a 24-tooth engine sprocket in combination with a 37-tooth clutch hub sprocket giving a primary ratio of 1.54 engine rotations for one rotation of the clutch hub and the transmission mainshaft.

Big Twins from 1994 to 2006 (except 06 Dynas) have 25-

tooth engine sprockets combined with a 36-tooth clutch sprocket giving a primary ratio of 1.44 engine rotations for 1 rotation of the clutch hub and the transmission mainshaft.

When performancing these 1994 and later models at Heavy Duty Cycles we will routinely retrofit the 25-tooth engine sprocket to a 24-size for better off the line get up and go. This will give a primary ratio of (24/36 = 1.5) 1.50:1.

Therefore, the engine will turn 1.5 times vs. 1.44 times, when using the stock 25-tooth engine sprocket for every clutch hub sprocket/transmission mainshaft revolution).

All 2006 Dyna, 2007 to 2009 Twin Cams have 34-tooth engine sprockets and a 46-tooth clutch hub sprocket giving a primary ratio of 1.35 engine rotations for one rotation of the clutch hub and therefore one rotation of the attached

transmission mainshaft that the hub sits on.

For those that only care about stoplight-to-stoplight drag racing, Screamin' Eagle (www.harley-davidson.com), borrows a page so to speak from BAKER Drivetrain who was first in developing a 21-tooth engine sprocket along with a new correct length primary chain that also adversely affects high rpm highway riding. To be fully accurate, early Harley FL police bikes from the Pan and Shovel era sometimes used 21-tooth engine sprockets for faster get up and go in city traffic although the engines just screamed on the highway. Highway police bikes of the era had engine sprockets up to 24-teeth for better highway speeds and lower engine rpms.

The Screaming Eagle offering (40331-02; 1994 & later FXST/Dyna: 40200-02 FLHT's 1994-2006) also available in its original form from BAKER (#155-56S FLT-FXR 1985-06 part

# 155-56L FXST/Dyna) changes the primary ratio from the stock 1.44:1 to 1.70:1. This new ratio causes the engine to spin at higher rpms at any speed with the 1.70:1 primary drive ratio that is 1.70 times vs. 1.44 times as a baseline that will help give the most torque and thus acceleration at the stoplight.

BAKER also has:

- 22-tooth (#156-56S FLT-FXR 1985-06 part # 156-56L FXST/Dyna),

- 23-tooth (#157-56S FLT-FXR 1985-06 part # 157-56L FXST/Dyna).

Kits come complete with correct length primary chains.

Two other integral components of the powertrain that

require much discussion are the clutch and the transmission.

## The Clutch

Harley-Davidson clutches have come a long way from crude, relatively inefficient bulky contraptions to lighter, stronger proficient units of torque transfer. Simple changes in mounting add strength but also eliminate some of the previous vibration.

### The Clutch Package as a Vibration Source

Think about it! The clutch package consisting of inner hub, steel and fiber plates, outer hub replete with starter ring gear and primary chain dual-row sprocket, pressure plate, diaphragm spring and assorted hardware hangs on the end of the transmission shaft at the inside rear of the primary system.

It whirls around at the differing rpms dictated by the primary reduction ratio.

A 2008 Twin Cam clutch package rotates one complete revolution for every 1.35 engine revolutions. Therefore at 4500 rpm, the clutch package is rotating (4500 ÷ 1.35 = 3334), 3334-revolutions per minute. Even at idle, (1000 ÷ 1.35 = 741), the clutch package rotates at 741 times-per-minute. This is a big weight to have spinning this fast, especially if out-of-balance. The clutch package is a vibration source along with the smaller but still weighty engine-compensator-sprocket assembly that the Factory and engine builders have ignored forever.

At Heavy Duty Cycles (www.heavydutycycles.com), we routinely add Mercury balancers to both the compensator and clutch packages when receiving customer vibration complaints.

# Clutch Mounting

Modern clutch hubs (1990-present) have straight splined-centers that mate with a splined outer diameter of the transmission mainshaft that extends out from the transmission and into the inside of the inner primary casing connecting the primary powertrain to the transmission powertrain. Earlier models before 1990 connect the clutch hub to the non-spline and tapered transmission mainshaft via a press fit keyway. This earlier mounting system is not as strong as a male multi-spline transmission shaft inserting into mating female spline clutch-hub-centers. The spline attachment arrangement eliminates potential vibration because the alignment of component parts is much squarer and thus more precise than the awkward skewing that many times occurs when fitting a male tapered shaft into a female clutch hub taper that locks

with a Woodruff key.

Vibration potential is evident at a glance, if watching running bikes in neutral, without their outer primary covers as we often do in our service department. The rotating Shovelhead and early Evolution keyway mounting clutch packages lack of rotational trueness is evident relative to a later spline mounting clutch assembly, The taper-mount clutch does not rotate perfectly true while the late Evolution and Twin Cam clutch spins comfortably and efficiently with nary a wobble.

This is another historical vibration reduction in present day Harley-Davidson's.

The Power Train

The clutch is part of the powertrain, which transfers torque

back from the engine through the shock absorbing compensator sprocket to the clutch sprocket via the attaching primary chain, into the transmission via the clutch to the transmission pulley and back to the rear pulley via a belt drive. A Harley-Davidson, like most vehicles and a lot of machinery requires a mechanism to interrupt this power flow to the rear wheel for stopping and starting without stalling the engine or damaging transmission gears.

## How a Clutch Operates

The center clutch hub fits onto the transmission mainshaft inside the rear of the primary on the right side of the Twin Cam, all Big Twins and Sportsters. The Big Twin and XL 1990 to present spline fitment is the strongest and has the truest alignment. A large clutch nut locks the inner hub in place. The outer clutch shell also known by the Harley colloquialisms as

the basket or drum can rotate freely on the inner clutch hub, with clutch engaged; clutch lever pulled in and bike not in motion. The two separate by a bearing or bearings. Twin Cams separate by one large bearing.

Now, a mechanism needs to hold the inner hub with the outer one to act as a single unit when wanting the bike to move forward.

Clutch Plates

The steel and fiber clutch plates that occupy space between the inner hub and outer shell perform this function. Clutch plates are round with a hollow portion in their centers. The Twin Cam thin steel plates have a serrated-pattern inner diameter that fit over mating outer diameter serrations on the center clutch hub. The thin fiber plates have pattern-serrations on their outer diameter that fit or lock into similar

inner diameter serrated clutch shells.

Some other H-D clutches are not serrated but have slotted outer diameter fibers like the 1973 to 1985 Ironheads. However, the XL mating steel plates has the serrated fitment arrangement for the steel plates to lock onto their clutch hub inner.

Big Twin 4-speeds and pre-1973 Sportsters, use clutch fingers that extend from the inner hub. The early Big Twins have fewer but fatter clutch plates while the early XL's use thinner plates like the later Harley-Davidson models. The 4-speed Big Twins fiber-clutch-plates have holes around their hollow centers that fit over the clutch fingers to lock them in place. The early XL's have holes in their steel plates that fit over the inner hub fingers. The Big Twin steel plates have slots on their outer diameters that mate with and lock to the

inner diameter of their outer clutch shell. The early Sportster fiber plates lock to the outer clutch shell. The clutch fiber plates are steel also but with heat-resistant friction material bonded to either side on the outer diameter surface.

The clutch plates, whether fiber or steel, inner locking or outer locking have loose fitment in order that they can move freely along serrations, slots or fingers, whatever the case may be. If denied free movement by warpage, too tight fitment or burring of mating parts, the clutch will not freely engage or disengage.

When the clutch disengages (handlebar clutch lever out), to put the bike in motion, the clutch plates lock or clamp together via friction material against steel. The effectiveness of a clutch is all about friction fiber material, how well it retains friction under extreme use and heat and the amount of friction surface area.

Old style Big Twin clutch fibers; three on pre-1967 models and four on 1968-1984 models are thick and have a wider band of friction material than later thinner plates. Later plates are better for performance because they are lighter, thinner and have up to eight fiber plates with narrower bands of friction material.

Clutch Parts

Pictured are a variety of clutch components going back to the 1957 Sportster and to the 1941 Big Twin. Beginning at the bottom left is a 1971 to 1984 Sportster clutch pack. This

includes the clutch shell with the electric start ring on its circumference. Hidden underneath is the triplex primary chain sprocket. The Sportster primary chain has three rows of links while the Big Twin including the Twin Cam has beefier 2-rows. The visible six lock-thread nuts holding the clutch pressure plate in place, thread onto the clutch fingers. Underneath the pressure plate on each of the six clutch fingers are six individual coiled springs surrounded by interspersed clutch fibers and clutch steels. The bearing in the center of the pressure plate is the clutch throwout bearing. Clutch plates are round with a hollow portion in their centers. The Twin Cam thin steel plates have a serrated-pattern inner diameter that fit over mating outer diameter serrations on the center clutch hub like that shown top right. The thin fiber plates have pattern-serrations on their outer diameter that fit or lock into similar inner diameter serrated clutch shells. Some other H-D clutches are not serrated but have slotted outer diameter fibers like the 1957 to 1985 Sportsters. However, the XL mating steel plates has the serrated fitment arrangement for

the steel plates to lock onto their clutch hub inner. Moving up clockwise is a 1957 to 1970 XL clutch hub or basket. The narrow slots on the inside of the outer hub are about the same size as those on the clutch fibers, which can move back and forth within the confines of each pair of the hub slots. The early hub is kick-start only since the Sportster did not get an electric start until circa 1973. The sprocket triplex sprocket rows of primary chain drive teeth are therefore visible. Next and above, is a Sportster diaphragm clutch spring; Big Twin including Twin Cam diaphragm springs have shorter, stubbier inside appendages. A diaphragm spring is a flat piece of sprung steel that rises towards its center. A flat pressure plate forces or releases against the diaphragm spring to activate or deactivate pressure. Some-year, rotating clutch-diaphragm-spring pressure plates can adjust profile for more or less spring pressure. Consult your manual for all clutch adjustments. Earlier Big Twins and Sportsters use a bunch of coiled springs for pressure clamping the clutch pressure plate against the alternating fiber and steel clutch plates. The XL

uses 6-springs while the Big Twin uses ten. The fiber clutch plate in the top left hand corner has external slots that lock into the outer clutch shell/hub/basket. The clutch fiber plates are steel also but with heat-resistant friction material bonded to either side on the outer diameter surface. The clutch plates, whether fiber or steel, inner locking or outer locking have loose fitment in order that they can move freely along serrations, slots or fingers, whatever the case may be. If denied free movement by warpage, too tight fitment or burring of mating parts, the clutch will not freely engage or disengage. When the clutch disengages (handlebar clutch lever out), to put the bike in motion, the clutch plates lock or clamp together via friction material against steel. The effectiveness of a clutch is all about friction fiber material, how well it retains friction under extreme use and heat and the amount of friction surface area. Next in the top right corner is a clutch steel plate. The Twin Cam thin steel plates have a serrated-pattern inner diameter that fit over mating outer diameter serrations on the center clutch hub. The thin

fiber plates have pattern-serrations on their outer diameter that fit or lock into similar inner diameter serrated clutch shells. Next is a Big Twin 1941 to circa-1984 clutch fiber plate. Big Twin 4-speeds and pre-1973 Sportsters, use clutch fingers that extend from the inner hub. The early Big Twins have fewer but fatter clutch plates while the early XL's use thinner plates like the later Harley-Davidson models. The 4-speed Big Twins fiber-clutch-plates have holes around their hollow centers that fit over the clutch fingers to lock them in place. The early XL's have holes in their steel plates that fit over the inner hub fingers. The early Big Twin steel plates have slots on their outer diameters that mate with and lock to the inner diameter of their outer clutch shell. Shown in the middle is a Big Twin 1941 to 1984 clutch finger assembly. Second from the right side bottom is a modern aluminum center-clutch-hub with thick serrated nubs around its outer diameter to mate with the clutch steel plates. Lastly, on the bottom right is a modern outer clutch basket with the ring gear hiding the primary dual-rows of primary sprocket teeth.

The locking serrations for the clutch fiber plates are on the inner diameter of the clutch shell. The 2006 Dyna and 2007 to present clutch shell (not shown) has the primary sprocket teeth move to the front of the ring gear on these newest H-D Twin Cam clutch shells.

Engaging and Disengaging the Clutch Plates

The mechanism for locking the clutch plates together is a spring(s) activated pressure plate.

Unlocking the plates or engaging the clutch achieves by pulling the handlebar clutch lever in; or on very old Harley's, stepping down on the foot clutch pedal sometimes known as a suicide clutch when the return spring is missing.

Later H-D clutches, circa 1986 to present Sportsters and circa 1985 to present Big Twins, use a diaphragm spring that

does not look like a traditional coiled spring at all. The clutch diaphragm spring operates on the same principles as the compensator diaphragm springs but opposite as one applies pressure while the others absorb pressure power pulses. The internal diaphragm clutch-spring pressure plate locks the clutch plates together via friction. This transfers engine power into the transmission for further gear reductions. Then power transfers onto the rear wheel via the secondary ratios of the transmission output pulley that moves as one with the output mainshaft final gear. Finally, the power moves back to the rear wheel pulley that bolts to the rear wheel.

From a performance and efficiency perspective, the increased number of plates used in the wet clutch provides greater clutch surface contact, transferring through a higher percentage of the horsepower and torque. The technical vernacular is that there is less frictional loss of horsepower.

Clutch Spring(s)

A diaphragm spring is a flat piece of sprung steel that rises towards its center. A flat pressure plate forces or releases against the diaphragm spring to activate or deactivate pressure. Rotating the clutch-diaphragm-spring pressure plate can adjust profile for more or less spring pressure. Consult your manual for all clutch adjustments.

Earlier Big Twins and Sportsters use a bunch of coiled springs for pressure clamping the clutch pressure plate against the alternating fiber and steel clutch plates. The XL uses 6-springs while the Big Twin uses ten.

Over the years, engaging the clutch involves three methodologies, stepping on a foot clutch on the left side of the bike by the front floorboard, hydraulically actuating the clutch via a handlebar clutch lever that uses hydraulic fluid and lines and the most common method, a clutch handlebar lever connecting to a cable. Pulling in the clutch lever (engagement), or letting it out (clutch disengagement), moves an actuator via mechanical steel rods, hydraulics or by a cable on the outer right side of the transmission opposite to the left side primary. The actuator is a ramp mechanism on later Harley's and a lever apparatus on earlier ones. Big Twins including Twin Cams 1990 to present utilize two connecting plates that separate by three single small roller ball bearings. The two mating actuator plates have a pocket for the ball bearings and ramps they can move up when the cable pulls one plate sideways against the other, thereby making the mechanism fatter or thicker.

This activation by pulling the clutch lever in pushes the actuator against a clutch rod or series of rods that fit inside the center of the transmission mainshaft. The clutch rod(s) extend over against the clutch pressure plate inside the left side primary. The clutch rod(s) movement pushes against the clutch pressure plate compressing the diaphragm spring or coiled clutch springs as the case may be.

This action pushes the pressure plate away from the clutch plates allowing them free engagement by unclamping or unlocking the friction against steel. This stops any transference of power back to the rear wheel. Conversely, when disengaging the clutch by letting the lever out, the clutch pressure plate now pushes against the plates via spring pressure clamping them together and transferring power back into the transmission through to the back wheel.

Clutch Cable Ease

Reduced clutch cable effort follows the design change in the 2006 Dynas. The reduced-effort clutch lever enables a 10% reduction for Softails and 7% for FLHT touring models.

Let us not leave the Sportster out, which has an 8% reduction.

## The Clutch Interlock Switch

There is a new clutch interlock switch specifically designed for new-rider safety and something us get-up-and-go, experienced riders have mechanically been doing forever. Many of us leave the bike in first gear when stopped. We turn the ignition on, pull in the clutch lever to engage the clutch and press the starter button and off we go.

This interlock button switch activates via the clutch lever position. It locates in the clutch-lever handlebar control. It is just behind an appendage of the clutch lever itself beside the clutch-cable attachment pin to the handlebar control piece. In the sometimes confusing Harley vernacular, the Factory describes this as an “or” function with the neutral switch. The interlock switch will verify if the bike is in neutral or not before permitting starting. If in neutral, the starter will function when pushing the e-start button in. If not in neutral, the bike will not start until pulling in the clutch lever thus engaging the clutch.

## How to Wreck a Clutch

Always follow the manual’s instructions of clutch adjustment because improper adjustment particularly on the slip side as opposed to the dragging side is the leading clutch

killer.

Slipping where the plates do not fully clamp together is worse than a dragging clutch where the plates clamp partially together when they should be free moving. A slippage example is grabbing a handful of throttle but the bike does not motivate forward as well as it should.

A clutch drag example is at a stoplight, in gear with the lever pulled all the way in but the bike still wants to creep or move forward. The clutch lever should have minimum one-eighth-inch free movement to prevent continuous clutch actuation pressure. No free movement is the equivalent of having the lever partially pulled in all the time. This burns out clutches.

Always adjust the clutch actuator at ambient room

temperature. Heat causes expansion (clutch slippage), and cold causes contraction (clutch drag), which affects proper clutch adjustment. Start by collapsing the clutch cable adjusters, which are the jam nuts and threaded extension portion of the outer cable usually protected from dirt by a rubber boot. This takes any pressure off the actuator. Remove the primary clutch derby. In the center of the inner clutch hub/pressure plate, area will be the clutch-actuator-adjuster screw. Loosen the lock nut. Eliminate all clutch rod free endplay by screwing the adjuster carefully in until it bottoms, which means the pushrod is against the actuator plate on the right side of the transmission. The actuator adjusting screw is in the primary on the left side of the transmission. Back off the adjuster screw at least one-half turn but no more than one full turn. Tighten the lock nut and replace derby cover. Pull in the clutch lever a few times to seat the actuator plate ball bearings. Now readjust the clutch cable to one-eighth-inch free play.

Adjust the diaphragm spring or coiled springs depending on year as per the manual. I will not go into this, as there is art and practical experience to adjusting the springs even with the manual. Spring(s) do not always have accurate equal pressure; pressure plates are not always true and flat; steel plates sometimes warp a little with the heat and finally, friction discs wear altering the adjustments.

Therefore, do not be afraid of practicing this adjustment and treat your work as a learning experience. Adjusting to the least pressure without causing slip makes for the smoothest riding experience. However, there is nothing wrong with a little drag for stock and mildly modified bikes.

Sometimes a little slippage will assist powerful engine torque transference with motorcycle handling control.

Primary Fluid is not a Clutch Plate Lubricant

Many think modern clutches are wet, which they are, but not for the purpose of clutch plate lubrication, since they will work wet or dry. Primary lubricant is necessary for primary chain and related parts lubrication, for sure. However, the primary fluid is very necessary as a clutch plate coolant.

The 4-speed clutches in all Big Twins and Sportsters up to 1972 are dry clutches. They will not tolerate any lubricant whatsoever. If oil or primary lubricant impregnates their fiber plates, extreme slippage will occur.

## Finding a Difficult Neutral

The neutral gear allows standing with the engine running also but is awkward for the purpose of stopping and starting without engaging and disengaging the power transfer.

Sometimes H-D and some aftermarket transmissions have a difficult time engaging neutral. First, ensure the cause is not the clutch dragging a little. Clutch drag occurs when the clutch is still active even with the clutch lever pulled in while in gear. If at a stoplight, clutch drag will cause the bike to creep forward with the lever engaged.

After checking this adjustment and finding neutral is still problematic, pull the clutch lever in while in first gear and place pressure upwards on the gear selector to pop the bike into neutral. Partially, let the clutch lever out. Many times the selector will slip into neutral. If still difficult, increase the rpm a little while applying light pressure.

Another technique is to up shift into neutral from first gear while the bike is still rolling. Also, try this rolling shift from second down into neutral. Many times, this neutral difficulty

is a result of too much free play in the clutch cable or the clutch actuator.

If the primary chain is too tight, a stubborn neutral may result because of chain pressure pulling on the outer clutch shell sprocket. Adjust the primary chain while cold in the tightest spot as chains wear unevenly. Adjusting on a loose spot will make the tighter spots way too taut.

Adjusting clutch spring tension too tight may help in a street race but may also make finding neutral difficult.

If the transmission is still being persnickety, the detent arm fitment inside the transmission shifting mechanism may need adjustment or perhaps a troublesome burr ground off.

## Speed Shifting

## Clutch Function

What exactly does a clutch do? The clutch is an uncomplicated way to interrupt power transfer without damaging parts or stalling the engine for the purpose of stopping and smooth acceleration when proceeding forward again.

## Shifting without Engaging the Clutch

While some transmissions can shift during some rpm ranges without a clutch, all the Harley-Davidson transmissions need a clutch for at least some of the gears during certain rpm ranges.

I quite often ride, shifting gears smoothly without engaging the clutch (clutch lever out-bike in motion). I have

been doing it so long that it is like second nature. I currently ride my 2007 Streetglide like this. I place light but firm pressure on the gearshift with my foot. If the rpm is right, my bike shifts easily into the next gear, up, or down without undue wear on any parts.

The key is not to force the lever when there is resistance. It will not want to go into the next gear anyway. In these cases, alter the rpm a little; usually increasing it when up shifting and the gear will usually pop in as if using the clutch. However, I cannot do this in every gear up and down. Well, I guess I could if practicing some more. I usually do this stuff because I will be eating and have my clutch hand full of food, be cold and have my hand in a warmer place or it might be sheer laziness.

Speed shifting is shifting gears without the clutch to save

time in a race. It saves time because the rider does not let rpms drop, pull in the clutch, shift the transmission, and let the clutch out again. Efficient speed shifting will occur in an rpm comfort zone for an effortless shift but also in an rpm range that will allow the bike to take advantage of the power band. Therefore, the sweet spot must coincide with power band advantage.

Although speed shifting is more aggressive than my lazy shifting, the same principles apply.

- Force does not work well whereas skill does.

- Force will wreck parts whereas skill will not.

- Force will not win the race whereas skill will.

Speed shifting, done correctly will not blow a transmission

but the process involves practice and a familiarity with the motorcycle's characteristics since gear spacing, shift detent and clutch adjustment will vary from bike to bike.

## Performance Clutches, Springs, and Pressure Plates

Ever since, I started riding decades ago on one of my first Harley's, a 1966 FL Shovelhead, the first upgrade was always clutch fibers, a Barnett clutch to be specific.

### 4-Speed Clutch Upgrades

Barnett Performance Products (www.barnettclutches.com) was the first to introduce the extra plate clutch kit, part number 304-30-10005, for 1941-1968 Big Twins. Back in the day, this was a very necessary addition for the performance enthusiast.

I had no money back then, so I would replace, the Electraglide clutch springs with stiffer ones from the Panhead. Barnett, supplies these springs to this day with their own performance upgrades; 1941-1967 clutch springs (10), part number 501-66-10041 give 50 lbs pressure at 1 ⅛" compression and are orange in color for easy identification. Blue colored springs (501-32-10068), for 1968-1983 have a weaker pressure of 40 lbs pressure at 1 ⅛" while their ten performance grey colored springs (501-90-10069) apply an authoritative 68 lbs pressure at 1 ⅛" compression.

Another trick is to add an extra fiber plate and a steel plate to the 1941-1967 3-plate clutch pack and turn it into a 4-plate clutch.

The thin stamped steel-pressure-plate always warped,

consequently never applying equal pressure. The final clutch upgrade, possible back then was to swap the stock pressure plate with a thick aluminum aftermarket one that did not flex and ensured even pressure. All this in the day, when the major aftermarket company Dixie Distributing sold mostly pistons and bottom end bearing rollers for rebuilds. AEE and the California chopper companies that marketed super long springer front ends, spool hub front wheels without brakes, and coffin gas tanks came a little later. There was not much available for a Harley-Davidson relative to today.

Barnett was king. This venerable aftermarket company began in 1948 and is still a major aftermarket player present day with their traditional quality steel and fiber plates, clutch springs, throttle, idle, and clutch cables. Barnett's excellent Scorpion clutch packs are very popular present day.

Asbestos was the choice of friction material but its carcinogenic qualities outlawed its use. Modern safer

materials like Kevlar and Carbon Fiber supersede materials of old.

Ram Jett

I forgot to mention another wonderful aftermarket product for these primitive clutch packs if I may go back to the 1941-1983 4-speed clutches for one last comment.

An annoying and counterproductive problem is that when the clutch engages, the outer clutch shell will ride outwards on its bearings partially separating from the inner clutch finger hub. This reduces actuator-clutch-rod travel. The Ram Jett clutch retainer saves the day. This simple and inexpensive item installs easily. It prevents clutch shell walkout.

I could not find a website address for Ram Jett but it is available from many aftermarket distributers.

Barnett Twin Cam Clutches

Fitment for the following parts and kits is for 1998-1999 Evolution models and all Twin Cams 1999 to present.

- The Kevlar clutch, (302-30-10013K), and the Carbon Fiber clutch (302-30-20013CF), use the latest and toughest materials with which, many readers are familiar. Some refer to Kevlar as bullet proof but Carbon Fiber is even stronger.

- The extra plate Kevlar (307-30-10013K) and Carbon Fiber (307-30-20013CF), clutches are a more serious option.

- A heavy-duty diaphragm clutch spring (502-00-01008), adds to clamping force.

- Finally, a Barnett conversion kit (511-30-10004) includes a billet aluminum-pressure-plate and two sets of six coil springs.

I have briefly covered the early Big Twin clutches and the latest Twin Cam ones but

Barnett and other suppliers like Alto, whose web address I could not find, supply quality clutches, steels, fibers, springs and pressure plates for all the Harley-Davidson's in between. These include Big Twins 1985-1989 and 1990-1997 as well as Sportster pre-1972, 1973-1985, 1986-1990, and 1990-1997. In the earlier years, XL and Big Twin clutches were completely different designs but in later years, both use the same clutch fibers and steels with some other common parts.

The Scorpion Clutch

Scorpion Billet Clutches have a 100% increase in clutch surface area relative to the stock Twin Cam clutch. It supplies with a CNC machined billet aluminum-pressure-plate, friction plates, steel plates, a steel inner hub, and additional sets of coil springs for personal tuning needs and wants. The Scorpion is versatile and can run wet or dry in an open or enclosed primary.

TPP Variable Pressure Clutch

Another interesting and functional Barnett performance clutch product is the TPP Variable Pressure Clutch. The TPP Variable Pressure unit provides up to 60% more clutch pack compression pressure. This enables the use of a lighter diaphragm clutch spring for up to 50% easier clutch lever effort with no loss of performance.

## Centrifugal Lock-up Clutches

Once the domain of the drag strip, centrifugal activating lock-up clutch heads enhance the clamping force of the coil clutch springs to apply more pressure to the clutch plate interaction as rpm's increase. It is interesting to note that most of these lock-up clutches prefer the old-technology independent coiled springs instead of the modern diaphragm spring.

A lock-up clutch usually uses weighted levers that will pivot outwards with the increasing centrifugal force of increasing rpms. This escalating force compels the weightless ends of the levers inward against the clutch pressure plate adding more pressure as rpm climbs and decreasing the pressure as rpm decreases.

Clutches depend on spring pressure to clamp the fiber and steel clutch plates together to improve their grip on each other. A problem with increasing spring pressure, whether they are coil springs and especially a diaphragm one in traditional clutches is that lever-pull becomes harder sometimes very harder. This is cool, if one is on a weightlifting regimen to develop a single Popeye forearm but otherwise is a distracting and muscle sore experience.

Lock-up clutches do not increase lever-pull. In fact, they may use lighter springs than a traditional clutch. The CNC machined, 6061-T6-billet aluminum Barnett Scorpion centrifugal clutch (608-30-20098), progressively boosts spring pressure as the rpm increases. It adjusts easily by adding or subtracting supplied weights on the six pressure arms.

There are other types of actuating lock-up clutches. One methodology uses individual steel ball bearings that move outwards on ramps to increase clamping force, once again by

the Principle of Centrifugal Force. These wonderful physics principles pervade every aspect of motorcycle operation and especially performance applications. I never thought I would refer to my most hated subject in school with such affection. It is my favorite subject now; I admit it, I am a sick-prick.

## Automatic Centrifugal Lock-up Clutch

Rivera Engineering (www.riveraengineering.com) uses go-kart centrifugal technology with their lock-up clutch. This eliminates the need for lever-pull. Sometimes the lever may make shifting a little smoother by pulling it in a little with boot pressure on the shift lever. This Centri-Force centrifugal clutch applies pressure between 1100 and 1500 rpms with an adjustable stall speed. The clutch handle stays against the handgrip at idle, but goes to its normal standoff position when accelerating. This clutch can shift normally, or shift without

touching the clutch lever using the power-off-technique.

## More Excellent Clutches

There are many other great clutches out there including Bandit and S&S Cycle's performance clutch pack. For so many years, performance clutching restricted to upgrading fiber compounds, spring pressure, and pressure plates. Now, there is a plethora of new transfer technology for Harley-Davidson application available to the performance enthusiast.

## Gear Ratios

Mismanagement of horsepower and torque gains often occur because of a general lack of knowledge about gear ratios. Production of power at the engine transferring back to the rear wheel in usable amounts relative to engine rpm, mph speed, and load on the bike is the true key to a functional

motorcycle whether it is for comfortable touring, stoplight racing, and everything in between.

## Historical H-D Gear Ratios

I am going to spend seemingly much time on Big Twin Harley-Davidson's exploring gear ratios and their historical use.

The reader will discover the big differences in the 2006 Dynas, with more changes in the 2007 Big Twins in overall gear ratios. This is a big reason for the increased performance seat-of-the-pants feel of these new models.

The conundrum is that the increase in cubic inches does not offer any more horsepower combining with the new six-speed no-overdrive sixth gear.

An overdrive sixth gear lowers rpm in that final gear.

H-D achieves a similar objective in its 1:1 final ratio six-speed. Harley does this by incorporating a new primary drive-gear reduction ratio of 1.35:1 as opposed to the 2005 ratio of 1.44:1. The new ratio spins the engine a slower pace of 1.35 times for one revolution of the clutch package (primary ratio), as opposed to a faster 1.44 times on 1994-2006 models, except the 2006 Dyna or 1.54 times on its predecessor five-speeds. (1985-93).

In review, the primary gear ratio calculates by dividing the number of teeth on the clutch hub sprocket by the number of teeth on the engine sprocket. The primary drive, consisting of the engine sprocket and clutch basket sprocket and the gearbox's internal gear ratios, should design to best utilize engine power and torque.

The primary drive transfers engine power from the crankshaft to the clutch and the transmission. A primary chain connecting the smaller engine sprocket to the larger clutch sprocket on the front or back of the clutch shell depending on year that encases the steel and fiber clutch plates, accomplishes primary drive reduction.

Overall Gear Ratios

Further reduction occurs with different-sized gearing in the transmission, combining with the size-transmission pulley and finally the rear-wheel pulley.

Total reduction, also known as overall gear ratios, indicates the number of engine revolutions required to drive the rear wheel one revolution. For example, a 1993 FLHT

model engine turns through 3.37 revolutions in fifth gear in order to turn the rear wheel once through 360-degrees.

I am using domestic USA sprocket sizing for all examples used in this chapter. International models use smaller rear-wheel-pulleys for better highway cruising. Some European highways like the famed autobahns in Germany have no top-end speed limits at all.

The 2005 domestic USA models' engine turns 3.15 revolutions in fifth gear to turn the rear wheel one revolution. This difference is due to a change in sprocket sizes in the primary since the internal gear ratios remain constant and the transmission pulley at 32-teeth or, more correctly 32-nubs, and the rear pulley staying at 70-teeth/nubs are the same also.

As I have said, the earlier pre-1993 5-speed models have

24-tooth engine sprocket and 37-teeth on the clutch sprocket. The1994 and later five-speed models use 25-teeth on the engine and reduce the clutch sprocket to thirty-six teeth.

The engine sprocket has more of an impact on gear ratio than the clutch sprocket on a per-tooth basis, as the transmission sprocket has a bigger impact than the rear wheel one. When performancing these at Heavy Duty Cycles, we routinely retrofit, the 25-tooth engine sprocket to a 24-tooth one for better off the line get-up-and go. This gives a primary ratio of (24/36 = 1.5), 1.50:1 turning the engine 1.5 times vs. 1.44 times using the stock twenty-five-tooth engine sprocket for every clutch hub sprocket/transmission mainshaft revolution.

I mention these overall ratios to demonstrate that a major change occurred with the six-speed 2006 Dyna and all

subsequent 2007 and later Big Twins. All the 2007 to present Big Twins carried over the 2006 Dyna's six-speed and primary gear ratio changes. The 2006 Dyna engine turns over only 2.96 times per rear wheel revolution. The 2006-Dyna and all 2007-Big Twins have an unprecedented 34-tooth engine sprocket compared to what we previously considered to be a lazy twenty-five or better performance 24-tooth one in previous years. They also have a big change with the clutch sprocket from previous years' sizing of either a lazier thirty-six or better performance 37-teeth to a whopping forty-six.

The 2007 Big Twins turn over even less at 2.79 engine revolutions per one rear wheel revolution. Although they have the same oversized primary sprockets, the rear pulley reduces to 66-teeth in 2007-2008 from 68-teeth on the 2006 Dyna. In 2009, the ratio for all Twin Cams reverts back to the 2006 Dyna ratio.

Certainly, the 2007 gear ratios would not be practical without the increase to 96 cubic inches, a dramatic increase in torque and closely matched internal six-speed gear ratios. If there is a secret to the 2007 Twin Cams, then this it!

Gear ratio performancing will be crucial to big gains with these later models in future years. One key will be installing 2006 and earlier primary ratios and/or the pulleys that affect secondary ratios and will need substantial change for performancing goals. More realistically, present gear ratios will modify as the aftermarket develops new products. This needs to occur because the compensator parts from 1989 to 2006 are not interchangeable with 2006 Dynas and 2007 to present Big Twins. This is due to a change on the engine crank sprocket shaft splines. The sprocket shaft extends into the primary where the compensator engine sprocket fits. Earlier models have ten fat wide splines while the new bikes have

twenty-four thinner multi-splines.

Early builders of bigger-cube 2007 engine models noticed a measurable loss in horsepower comparing to the results if they had done the same upgrades to the 25/36 or the 24/37 primary reduction gear ratios in earlier Twin Cams. We are all attributing this to the drastic changes in primary reduction sprocket and clutch gear sizing. This also explains one major reason why the 2007 TC96 models do not have the expected increase in horsepower over the previous Twin Cam 88.

However, there is a dramatic and positive increase in torque with the new 2007 configurations.

## Secondary Drive Gear Ratios

The relative sizes of the transmission pulley/sprocket and

the rear wheel pulley or chain sprocket determine secondary drive ratios.

$$\text{Secondary gear ratio} = \frac{\text{Wheel pulley teeth}}{\text{Transmission pulley teeth}} \text{ X high gear ratio}$$

High gear ratio refers to the final gear ratio inside the transmission, which for all H-D stock four-five-six-speeds is 1:1.

Therefore:

- The FLH Shovelhead is (51/23x1.001 = 2.22), 2.22, which means the transmission sprocket and transmission mainshaft turn 2.22 times for 1 rotation of the rear wheel sprocket. To state the obvious, the rear wheel turns one time for each revolution of the wheel sprocket.

- The secondary gear ratio for all rubbermount five-speeds from 1985 to 2006, excepting the 2006 Dyna, is the same 2.19:1 since they all use a seventy-tooth rear wheel pulley and a thirty-two-tooth front transmission pulley. Thus, on these models the transmission pulley and transmission mainshaft turn 2.19 times for one rotation of the rear-wheel sprocket.

- The secondary gear ratio for the 2006 Dyna still uses a 32-tooth transmission pulley but a smaller 68-tooth rear-wheel pulley (68/32 x 1.001 = 2.13). This results in the transmission pulley and transmission mainshaft turning 2.13 times for one rotation of the rear-wheel sprocket.

- Next, the secondary gear ratio for all 2007 to 2008 Big Twins calculates by using the 32-tooth transmission pulley and the again smaller 66-tooth rear-wheel pulley (66/32 x

1.001 = 2.06), resulting in the transmission pulley and transmission mainshaft turning 2.06 times for one rotation of the rear-wheel sprocket.

- Finally, the secondary gear ratio for the 2009 Twin Cams still uses a 32-tooth transmission pulley but a larger 68-tooth rear-wheel pulley (68/32 x 1.001 = 2.13). This results in the transmission pulley and transmission mainshaft turning 2.13 times for one rotation of the rear-wheel sprocket.

- The 2009 models revert to 2006 Dyna secondary gear ratios.

The reader can see from the chart that both primary and secondary drive ratios are steadily decreasing and thus producing smaller final drive gear ratios over the years to the lowest levels ever with the 2007 to 2008 Big Twin.

Sprocket/Pulley Sizes

Primary: Secondary and Final Drive Ratios

32

32

34

The reason that 2006 Dynas and 2007 to present Big Twins have differing gear ratios is because of a two-tooth/nub reduction in the 2007 to present rear-wheel pulley. The 2009 Twin Cams revert to the 2006 final drive gear ratio by going up to a 68-nub rear pulley.

Final Drive Gear Ratios

I will now explain how to calculate the final drive gear ratios. Most riders and enthusiasts concern themselves with this ratio. The primary and secondary ratios determine the final drive ratio.

Final drive ratio = primary gear ratio x secondary gear ratio

Therefore:

- The 1973 Ironhead XL 1000cc final drive ratio is (1.74 x 2.43 = 4.23 or 4.23:1), which means the engine turns 4.23 times for each rotation of the rear wheel.

- The FLH final drive ratio is (1.61 x 2.32 = 3.74 or 3.74:1), which means the engine turns 3.74 times for each rotation of the rear wheel.

- The final drive ratio for all rubbermount five-speeds from 1985 to 1993 is (1.54 x 2.19 = 3.37 or 3.37:1), having the engine turn 3.37 times for each rotation of the rear wheel.

- The final ratio for 1994 to 2006 (excepting the 06 Dyna) is (1.44 x 2.19 = 3.15 or 315:1), turning the engine 3.15 times for one rotation of the rear wheel.

- The ratio for the 2006 Dyna is (1.35 x 2.13 = 2.88 or 2.88:1) so that the engine turns 2.88 times per rotation of the rear wheel.

- The final drive ratio for all 2007-2008 Big Twins is (1.35 x 2.06 =2.79 or 2.79:1). Thus, the engine turns 2.78 times for each rotation of the rear wheel.

- Lastly, the ratio for the 2009 Twin Cams reverts to the

2006 Dyna ratio is (1.35 x 2.13 = 2.88 or 2.88:1) so that the engine turns 2.88 times per rotation of the rear wheel because the rear pulley increases from 66-nubs to 68-nubs.

Changing the rear pulley size can easily modify the BAKER and S&S six-speed final drive ratios.

Comparing TC96 Break-in Procedure with the Past

I could not believe the ease of breaking in a new TC96 with the supercharged primary gear reduction in combination with the 6-speed transmission.

I have owned many Shovels over the years. The FLH models had higher gearing for adequate city riding. They allowed comfortable lower highway speeds for the era but not very high ones. My Shovels would not allow me to attain

main highway speeds during break-in periods for a new engine. I had to stick to secondary two-lane highways and suburban thoroughfares in order to keep the engine rpm to an acceptable level. Even when broken in, these bikes simply cannot keep up with the sustained highway speeds that the Evos effortlessly make.

Stock-geared Evos were much easier to break in than the Shovels in today's fast moving world. I could easily stay in highway slow lanes while putting the first all-important miles on them. Their more precise cylinder boring with benefit of torque plates along with eutectic pistons, and better ring design made much improvement over Shovel technology. This further combines with a superior and rigid enhanced cylinder-tie-down system. Improving gear ratios were beginning to show their benefits also. The break-in procedure involves gradually and alternately increasing, then decreasing engine rpm to allow mating parts like cylinder walls and piston rings to wear into each other. Once broken in, I have ridden many

Evos on cross-country trips at a steady 80 mph.

A Shovel can only reliably do this at 60 mph. I know that many will disagree out of loyalty to their Shovels. I am not attacking the Shovel, as it is one of my favorite bikes but the technology, metallurgy and parts precision is simply not up to sustained high speed riding. Hypoeutectic pistons without protective coatings, bored without benefit of torque plates fit sloppily into cylinders that have little supporting rigidity especially in the earlier Shovels. The 1981 to 1984 Shovels had benefit of some improvement changes like offset connecting rods, unleaded gas valve seats, improving ring design and better but not the best eutectic pistons as engineers made ready for Evolution production.

My 2006 FLHXI TC88 Street Glide and I wound up on a major highway with less than twenty miles on it. My friends

promised to stay at 60 mph or less, so I could gradually break in the engine. Soon, they forgot their promise, and we were going 65 mph and then 70 mph. It was a cool day, and the engine suffered no ill effects. When broken in, sustained speeds of 90-100 mph did not burden the bike at all.

My new 2007 FLHXI TC96 and I suffered the same fate as when breaking in the 2006 Streetglide. The difference was the very low temperatures. I knew the cold weather would allow me leeway in more speed. Once more, my friends began to leave me behind. I had to stay at close to 80 mph to catch up, but the rpm on the tachometer assured me I was barely within safe limits. Before the FLHXI was broken in I could tell this will be the fastest top-end Big Twin ever produced.

Soon after, I took my engine apart to hop it up. I was anxious to look at the cylinder walls and piston thrust surfaces to see if I was correct when going so fast during break-in. There was no damage or any wear period, and the bike was

breaking in just as Mother Harley would like. This is quite remarkable. This is primarily (no pun intended), due to the gear ratios married to the Cruise Drive six-speed transmission and the TC96 engine in a match made in heaven. It also has much to do with hypereutectic pistons with Teflon coatings, fit precisely into shorter rigid cylinders.

The above historical examples demonstrate in part, how progressively over time the primary and secondary with the resultant final drive gear ratios have improved. The above chart Sprocket/Pulley Sizes Primary: Secondary and Final Drive Ratios shows how the Factory has done this through lowering the final drive ratios over time.

The modern Harley can now hold its own on any highway including most European ones, excepting the fast lane on the autobahn. I have ridden the autobahn over twenty years. Even

the V-Rod has trouble keeping up in the fast lane.

Performance is all about increasing horsepower and torque. The second component, drive ratios, delivers this power in a way specific to the rider's needs and wants. Those who forget about or ignore primary, secondary, internal, and final drive ratios do so at their own peril. There is no quicker way to lose a race or have an uncomfortable touring motorcycle than with a wrong powerband for that purpose.

Gear ratios are the forgotten part of the horsepower and torque equation for delivering usable power to the rear wheel. Outside of the engine, gear ratios determine the powerband where peak horsepower develops and falls off. You as a rider want the powerband matching to the horsepower and torque of a particular engine and married to the motorcycle's function. Many mechanics and riders mismanage the

powerbands effective range. They mistakenly engineer peak horsepower beyond or out of the normal rpm capability of the engine. What is the point having peak power in the powerband at 6500 rpm when the engine may only be safe to take to 5500 rpm?

Normal riding is, give or take 500 rpm, from 2000 to 4000 rpm and this is where the powerband should be functional. Riders will want to alter the above general powerband range specifically to their riding styles. Those who ignore or do not understand primary, secondary, internal, and final drive ratios are at a big disadvantage. We need to manage the power transferring back to the rear wheel in usable amounts relative to engine rpm, mph speed, and load on the bike.

It is a time-honored tradition for H-D riders to change sprocket/pulley sizing in order to affect overall drive ratios for their riding style or sometimes just for the flavor of the day. In general, increasing a belt drive transmission-pulley on

a five-speed from the stock 32-teeth to a modest 34-tooth will decrease engine rpm by around six percent.

A concern with radical gear tooth changes resulting in lowering or raising an engine's rpm may take the engine out of its normal powerband speed. This may increase probability of pinging or spark knock under load and would really affect performance in specific area(s) if changing too much.

## Internal Transmission Gear Ratios

Internal transmission gear ratios have two main purposes.

- The first is obvious in that they transfer power back from the engine to the rear wheel when the clutch is disengaged (lever out).

- The second is to capitalize on the engine's powerband at differing rpm levels, speeds, and loads. The individual gear sets allow the rider to select a specific gear from four, five, or six choices depending on whether the transmission is a four- five- or 6-speed.

In other words, the rider is able to select a gear that is the best for the combination of rpm, mph, and the load or no load the bike is under at that time. For example, acceleration is a load situation whereas steady highway cruising is a no-load circumstance.

An internal gear ratio compares the number of times the clutch hub/mainshaft turns relative to one turn of the transmission pulley or sprocket, as the case may be, the higher the ratio, the lower the gear.

I quote BAKER Drivetrain for a compelling reason for the 1:1 final gear ratio, “Power flow through the transmission, the 1:1 gear is always the most efficient gear, typically 99+% efficient. Meaning, for every 100 ft/lbs of torque that goes into the transmission, 99ft/lbs of torque comes out of the transmission and goes to the rear wheel. To compare, the sixth gear in an overdrive unit is around 92% efficient. However, by going to a 1:1 sixth gear we lost the benefit of rpm reduction while cruising on the highway. We have ‘overdriven’ the primary drive to adjust the overall drive ratio by using a twenty-eight-tooth compensating sprocket in place of the stock twenty-four/twenty-five-tooth sprocket. The result is a six-speed that is exactly, ratio-wise, like the (BAKER) six-speed overdrive (OD6).” 1

The talented and innovative Bert Baker a former automotive transmission development engineer for one of the Big 3 creates a primary ratio of 1.29 when comparing with

1994 to 2006 five-speeds whose primary ratio is 1.44:1.

- Therefore, the BAKER DD6 has a primary ratio of 1.29 engine rotations for one rotation of the clutch hub and therefore one rotation of the attached transmission mainshaft that the hub sits on by using a twenty-eight-tooth engine sprocket and a thirty-six-tooth clutch hub sprocket.

The 1.29 primary ratio is calculated by dividing the 36-tooth clutch sprocket by the 28-tooth engine sprocket (36 ÷ 28 = 1.2857: rounds off to 1.29)

- The 1994 to 2006 five-speed Big Twins have a primary ratio of 1.44 engine rotations for one rotation of the clutch hub and the joined transmission mainshaft that the hub sits on by using a twenty-five-tooth engine sprocket and a thirty-six-tooth clutch hub sprocket.

# Internal Transmission Gear Ratios

I need to know the number of teeth on the mainshaft gear and its mating countershaft gear for the gear ratio I am going to calculate.

Final Mainshaft drive gear teeth X Mainshaft speed gear teeth

Final Countershaft drive gear teeth Countershaft speed gear teeth

The speed gear is the gear for which I am calculating the internal ratio.

For example the internal transmission ratio for a 5-speed gear is

$$41 \text{ X } 31 = 3.21$$

$$22 \quad 18$$

The resultant figure 3.21 means that there are 3.21 revolutions of the clutch sprocket for one revolution of the transmission pulley in gear. Direct drive 1:1 ratio in the final gear results in one revolution of the clutch sprocket for one revolution of the transmission pulley.

Harley-Davidson created an overdriven primary drive with its direct drive 1:1 ratio six-speed by increasing the compensating engine sprocket up to 34-teeth from the previous 25-tooth one and the number of clutch teeth to 46-teeth from the previous 36-tooth one, giving a 1.35 primary ratio.

• Therefore, 2006 Dynas and 2007 to present Twin Cams have a primary ratio of 1.35 engine rotations for one rotation of the clutch hub and attaching transmission mainshaft through using the new size primary sprockets.

Performance is all about increasing horsepower and torque. The second component, drive ratios, delivers power in a way specific to rider needs and wants.

Transmission Internal Gear Ratios

* BAKER has two available first gear ratios with the DD6 (3.77 or 3.45). The BAKER OD6 has three first-gear choices (3.21 or 2.94 or 2.82), two second-gear choices (2.21 or 2.08),

and two sixth-gear choices (0.86 overdrive ratio: 2776 rpm @ 75mph: or 0.80 overdrive ratio: 2583 rpm @ 75mph). BAKER also has an overdrive six-speed called the R ratio 2.75; 2.05; 1.55; 1.18; 1.00 and a sixth gear overdrive of 0.86)

** The BAKER DD7, Direct Drive 7-Speed is a direct replacement for the Cruise Drive Harley-Davidson factory 6-speed. The DD7 improves the Cruise Drive in many ways in addition to adding another gear. The DD7 has a shorter first gear for improving get-up-and-go. As a result, of the shorter first gear and differing gear ratios, a DD7-fitted bike will feel like it has enhanced engine work. Please see Chapter XVI: 2009 Model Changes for updates on internal gear ratios. The 2009 Softail and Dyna have the same internal gear ratios as the 2006 Dynas and 2007-2008 Twin Cams. 2009 touring models have updated internal gear ratios.

I will also summarize the benefits of an overdrive sixth gear like that found in the .089 Screamin' Eagle and .8931

Rev-Tech transmissions. The overdrive aftermarket six-speeds, which are mostly a .089 ratio, also coming in .086 and the radical .080 ratios, in their final gear, eliminate the 1:1 ratio disadvantages of a higher rpm in sixth gear if using stock primary and secondary drive ratios. There is still the same good initial take-off power and acceleration combining with a lower rpm and smoother highway riding in sixth gear with the added bonus of conserved fuel consumption. These work well with more cubic inches but this is not necessary with the .089 ratios.

My own experiences dictate the use of these .089 transmissions with performance engines and/or for bikes and their riders that do high-mile-per-day touring. If you do not fit into either category, there is no real need for one unless luxury is your game. In my opinion, in general, the overdrive transmissions work best in 2006 and earlier Big Twins with stock or close to stock gearing in the primary and rear drives.

The direct drive 1:1 transmissions come into their own when overdriving the primary, as with 2006 Dyna-2007 Big Twins or the BAKER DD6 that comes with a twenty-eight-tooth engine sprocket.

S&S uses a thirty-four-tooth transmission pulley with seventy-tooth rear pulley. All the transmissions we are talking about here are very cool when used with their respective primary ratios.

## USA Overall Gear Ratios

H-DI (H-D International) models generally have taller gearing for better highway touring because of increased speed limits. International models usually have smaller rear wheel pulleys than those in the USA

Overall gear ratios show the number of engine revolutions required to turn the rear wheel one 360-degree turn. For example, in first gear a 2007 model takes 9.31 engine revolutions to rotate the rear wheel once, while a 1999 Big Twin takes 10.81 engine revolutions to obtain the same goal in first gear. Overall gear ratios take into account the internal transmission ratios with the primary and secondary gear ratios, while the final drive ratio is a simple multiplication of the primary ratio times the secondary ratio.

*Except 2009 touring models.

## Performance Transmissions

The aftermarket has been producing reliable and efficient six-speed transmissions for close to a decade now, whether

they be full-sized gear transmissions made by BAKER Drivetrain or the first off-shore ones produced by Custom Chrome's performance line, Rev Tech (www.customchrome.com), with less beefy and lighter but very strong chromemoly gears with many newer manufacturers in between.

Offshore countries like Korea produce several variations of the above transmissions. The Harley rider should understand that reputable automotive transmission plants build many of these before conjuring up visions of cheap unreliable products. As testament to their quality, some offshore brands come with full replacement 3-year full warranties. Others have two-year warranties. The Rev Tech offshore overdrive six-speed comes with a limited warranty of five years/50,000 miles. Rev Tech used to have extended warranties on their six-speed builder's kits but have reduced the warranty period because of being unable to control the quality of the

installations.

After I wrote about offshore transmissions having better warranties than some American made products in American Iron Magazine, BAKER Drivetrain stepped up to the plate, fully confident in their American made quality products and offered a limited warranty of five years/50,000 miles for their excellent direct drive six-speed (DD6). Late in 2007, JIMS (www.jimsusa.com), announced that their American made transmissions, also of excellent quality, would now be backed with a six-year/60,000 mile warranty.

The plummeting price of aftermarket six- speeds in the last few years, dropping as much as sixty percent, is making them very affordable.

In general, Harley quality is top-notch. Nevertheless, has H-D ever offered a full-replacement three-year warranty? The

consumer is hard-pressed to get a comprehensive warranty of any type if buying Screamin' Eagle (www.harley-davidson.com), performance parts like a complete six-speed transmission gear set.

Transmission Parts Comparison

2005, 2006 Dyna and 2007 to Present Big Twins*

There are three dog rings in the H-D six-speed, two (first/second and fifth/sixth), with dogs on either side, or one pocket (third/fourth). The pocketed dog ring has no dogs or lugs but six indents on either side known as pockets. The advantage of these dog rings is that there is less inertia involved in moving them back and forth to engage with their

respective gears when compared with having to move whole gears to engage other whole gears, as in the five-speed. In order to make the dog rings even lighter with corresponding less resistance to movement known as inertia, they ride on guide hubs. The three guide hubs, one specifically for each dog ring is multi-splined inside and out. The inside splines rides on mating splines on their respective shaft, two on the countershaft and one on the mainshaft. The shift dog rings are also splined on their inside diameter to mate with the splines of the outside diameter of the guide hubs. Now the Cruise Drive lightweight shifting mechanism is nimble and has the least inertial resistance possible of any previous H-D transmission, a huge improvement over the bulky five-speeds.

*Except 2009 touring models, which have different internal gear ratios and thus different gears and gear sizing.

The BAKER 7-Speed TC96 Transmission

The BAKER DD7, Direct Drive 7-Speed (www.bakerdrivetrain.com) is a direct replacement of the Cruise Drive Harley-Davidson factory 6-speed improving it in addition to adding another gear.

The DD7 has a shorter first gear for improving get-up-and-go. As a result, of the shorter first gear and following gear ratios, the bike will feel like it has enhanced engine work. The BAKER DD7 transmission has the following ratios with comparative H-D Cruise Drive ratios in brackets: 3.76 (3.34), 2.75 (2.30), 2.06 (1.71), 1.55 (1.41), 1.27 (1.18), 1.10 (1.00); 7th (1.00).

I believe the shift quality is much better also.

In transmission design, a low mainshaft weight is always

desirable for smooth and quiet shifts. The factory mainshaft is an integral one-piece design that includes first, second, third, and fourth gears. The heavy rotating weight and the resultant high inertia of the 1-piece forged mainshaft creates the audible shift clunk in the factory 6-speed. The BAKER DD7 mainshaft is lightweight and only includes the small first gear as an integral part to reduce weight. Therefore, the shift clunk substantially minimizes.

Shift smoothness in the Factory 6-speed is a definite improvement over the predecessor 5-speed. The DD7 is a big improvement over the factory 6-speed because it incorporates a new linear roller ball detent as part of the supplied billet top cover. This linear roller ball detent is similar to the technology in first use in the BAKER TorqueBox transmissions. The BAKER, DD7 shift drum design helps the rider find neutral every time.

The Factory 6-speed has straight cut first and fifth gears,

which yields gear noise in these gears most noticeably in fifth. The BAKER DD7 has a full set of helical gears for quiet operation in every gear.

The BAKER DD7 design features include:

- Ease of Finding Neutral
- Smooth Shifting
- Clunk and jerky shift reduction
- Utilizes ball bearing linear detent in the top cover that was an original design for the BAKER Torque Box
- The DD7 Builder's Kits will come ready to slide in the

bike with no stock 6-speed case modifications, shimming of the gear set or adjustments.

- The 7th main drive gear bearing and seal are included to replace the stock sixth gear components.

- Superior hardware fasteners, ARP (Automotive Racing Products) 12pt bolts for trap door, top cover and the stock side cover.

- Includes the BAKER speedo re-calibration

- Utilizes the BAKER F6F Door Bearing Retainer Plate

- DD7 trap door bearings are 28% wider than stock.

Comparing Six-speed Transmissions

The H-D six-speed transmission will offer surprises when changing external gear ratios to gain more horsepower. External ratios may need modification to coexist in a positive way with the Factory 6-speed when performancing the engines. Combining these with a lot more powerful engines will involve creative sprocket/pulley changes to achieve desired results.

The stock six-speed has a 1:1 internal ratio in sixth gear while the Screamin' Eagle six-speed has an overdrive sixth gear with a ratio of 0.89:1. RevTech has offered a .8931 overdrive sixth gear ratio since around 1999. The .089 overdrive transmission will lower rpm about 360 rpm in sixth gear. However, it is not for use in the 2006 Dyna or all 2007 Big Twins. Differing versions for 1985 to 1989 and 1990 to 2006 fit 1985-2006 Big Twins. With the H-D, 1:1 ratio four-

and five-speeds, it is important that gear ratios be kept within bounds, known as the powerband, to ensure good start-off power along with acceptable highway cruising rpm on an all-purpose street Harley's.

A disadvantage of overdrive transmissions is that they require four gears to interact in sixth gear, and strangely, so does the H-D, 1:1 ratio direct drive six-speed.

An advantage of the BAKER direct drive is that a higher percentage of torque transfers through sixth gear relative to an overdrive transmission and the H-D direct drive since only two gears are involved at this point.

The .089 overdrive aftermarket six-speeds work well with more cubic inches but there is some torque loss because four gears are required to transfer sixth gear torque.

Even though Harley-Davidson has done an initially satisfactory job with its 2007 transmission combining with primary drive gear ratios, I did not quite understand why

H-D did not use an overdrive sixth. I say hesitantly, because there are changes that could make it better. I will demonstrate as this chapter progresses that the engineers could have further refined the six-speed design to reduce the transfer of torque power loss as well as addressing a nagging noise issue.

I asked Bert Baker of BAKER Drivetrain what he thought; “Overdrive six-speeds are inherently noisy in sixth because it is virtually impossible to make a large gear on the mainshaft drive a small pinion on the countershaft quietly, which is the case on the sixth gear pair in six-speed overdrives. High-quality transmission makers like BAKER spend a lot of money

on grinding the flanks of the gear teeth after heat-treating to minimize these noises, whereas some lower-quality transmission makers do not. I feel that this inherent noise with six-speed overdrive designs is the reason why Harley did not opt for this route." 2

Why would H-D build an overdrive that is intrinsically noisier when the whole goal of the helical gears was to appease the noise cops? I feel that the H-D six-speed transmission will improve further in coming years. The H-D six-speed may even be in for a redesign since Harley riders are not as sensitive to noise as the environmental folks are.

The Factory dampens these new noises with a rubber-mounted rear-wheel-drive-pulley beginning in 2008 on FLT models. However, it looks as if the 2-row extended race wheel bearing (part number 9285), supporting the pulley is having some warranty-wear problems. The pulley is redesigned in 2009.

Bert continues, “Being that the H-D transmission is direct drive in sixth gear, like our DD6, there does not ‘need’ to be four meshed gears. Just sixth main and sixth counter meshed. The H-D design drives three gears and has a fourth along for the ride. It does not utilize the advantages that a 1:1 drive ratio can give you. In a traditional overdrive six-speed the four meshed gears are needed to reduce the drive ratio below the 1:1 threshold to the (.89) .86:1 or .80:1 ratio depending on the transmission. Not so with (our) Direct Drive.” 3

The .086 and .080 final overdrive ratio six-speeds should combine with progressively bigger-inch bikes to achieve a satisfactory all-around powerband.

The H-D, BAKER DD6, DD7 and the S&S Cycle, Inc. (www.sscycle.com), 1:1 ratio six-speeds attain their goals

through another method: changing the primary or secondary sprocket/pulley ratios. Both the BAKER and the H-D transmissions give the response of an overdrive in sixth gear even though they have a 1:1 final sixth gear ratio. Harley overcame the overdrive issue by supercharging the primary gear ratio.

The conservative S&S 1:1 six-speed uses a 34-tooth pulley up from the stock 32-nub one. BAKER's DD6 (Direct Drive 6) 1:1 comes with a previously huge 28-tooth engine sprocket and longer primary chain. BAKER can also provide a sportier twenty-seven-tooth compensating sprocket for use with his Direct Drive 6-speed. When using stock pre-2006 clutch, transmission, and rear wheel pulley/sprockets, Bert Baker's DD6 allows the rider to mimic the 2006 Dyna- 2007 Big Twins gear ratios. BAKER's final drive ratio of 2.83:1 is right in the middle of the 2006 Dyna's 2.88:1 and the 2007 Big Twin final drive ratio of 2.78:1. The S&S six-speed comes in at 2.97:1,

lower than the 1985-93 Big Twins but higher than the 1995-2006 (except the 2006 Dyna), models.

The S&S transmission has a different goal than the other manufacturers in keeping with turndown ratios, which I will discuss later.

Sprocket Change Limitations

The rudimentary external rpm changes are not practical in every case. Nor are changes made to every sprocket in everyday usage. Fixed sprockets on different years and space limitations affect the amount of change on most years, although these are not hindrances to achieving changes in gear ratio goals in recent years. On modern Harley's, modifications are possible only on the engine, transmission, and rear sprocket/pulleys. The clutch sprocket remains fixed on the majority of years. They can change by one tooth from

1990 to 2006 by swapping the clutch package and some starter parts, but this is hardly practical.

## Early Sprocket and Pulley Changes

Primaries earlier than 2006 have no problem in accommodating wild off the line twenty-one or 22-tooth sprockets to the more lethargic 24 or 25-tooth stock engine sprockets. Rear-belt-drive models have a choice of going up or down a few nubs on the transmission pulley. Rear pulleys are common in a range from sixty-one to 70-nubs. There are correspondingly different-length H-D and aftermarket belts to accommodate them. However, there is no hindrance whatsoever in making whatever changes we may have in mind for a particular sized engine and type of riding combined with rider and bike weight.

There were restrictions in the choice of ratios in yesteryear models before the advent of the rear disc brake in 1973. The reason was the lack of aftermarket parts for the compensator engine sprockets and by the fixed non-changeable clutch and rear 51-tooth chain sprockets. They were integral with the clutch shell and rear brake drum respectively.

However, twenty-one and 22-tooth engine sprockets were readily available from the Police FL models and 23-tooth ones available from later FLH citizen Big Twins. Different-length primary chains were also more difficult to get unless eliminating the endless chain by splitting it, cutting, or adding to the length and rejoining, using a master link(s). BAKER resolves these issues for modern riders with its sprocket and primary chain kit combinations.

Most changes made during the old days were to the

transmission sprocket, which came in every size from nineteen to 25 teeth, the 24-tooth aftermarket sprocket being the most popular. I liked the 25-tooth sprockets just as much but interference with the starter housing after 1965 became as issue with a loose or unevenly stretched rear chain. Inattention causes chain maladjustment. Chains also usually have loose spots because of the uneven stretching characteristics of drive chain. These restrictions combining with a four-speed transmission with a 1:1 ratio fourth gear limits our choices along with the anemically powered Shovel and predecessor engines. I use this as a reference to the liberal changes allowed today with five and six-speeds. There is a wide array available from both Harley and the aftermarket for the non-fixed sprocket/pulleys excepting the clutch sprocket.

## Evolving Rear Belts

The 2006 Dyna uses the thinnest belt 1 ⅛” (40015-00, 133 tooth) to date transferring the power back to the rear wheel.

Rear-drive belts have been continually getting thinner widths since their introduction decades ago. Just how strong can a Kevlar belt be?

Harley has chosen to introduce a new, yet thinner width drive belt in 2007 with its biggest-ever TC96 engine. I must say I was nervous when looking at my 2007 belt but I will explain and will alleviate any concerns a little further on.

Increasing Engine Size

The Factory made the change knowing that it's Screamin' Eagle kit 3 ¾" bore to 3 ⅞" bore kit will increase the 96c.i engine to 103 cubic inches. Yes, a mechanic can use original TC96 cylinders and bore them out rather than purchase new cylinders since 103 S.E. cylinders are bored out 96 cylinders. The Factory is also offering a kit to take the 96 out to 110 cubes and another to 113c.i, so what gives with a thinner

width drive belt only one-inch wide?

The new 96c.i engines do not offer any substantial increase in horsepower over the TC88s because of the overdriven primary reduction ratio but give gobs more torque.

- The FLHT touring models engine torque increases to 93 ft-lbs @ 3500 rpm.

- The Dyna models engine torque increases to 92 ft-lbs @ 3000 rpm.

- The Softail models engine torque increases to 90ft- lbs @ 2750 rpm.

These measurements are at the engine and not the rear

wheel. Loss is anywhere between twelve and twenty percent as the power transfers back. The higher losses result from such factors as chain and belt tensions, clutch adjustment, tire pressure, and motorcycle misalignment, etc.

Touring models have bigger capacity mufflers than Dynas or Softails and, therefore, slightly more horsepower and torque.

Beta Softail models have less torque than the Dynas because of internal loss of horsepower due to the energy required to drive the engine balancers.

More torque requires a stronger drivetrain.

New Generation Belts

Original 1 ⅝”-wide belts used Aramid fibers colloquially

known as the bulletproof Kevlar.

To demonstrate their strength, Harley published a picture of an FLHT hanging in the air supported by one thin Aramid fiber.

The X7N is the name for new final 1”-wide belt. It composes of the legendary second-generation carbon fiber. The X7N utilizes thicker fabric for increased tooth nub strength and durability. The polyurethane outer core body opposite the nubs is the same material used in previous width belts. The tensile cord in the middle is comprised of the new carbon fiber space-age materials filtering down to everyday usage is decreasing previously necessary bulk in all types of applications in everyday life.

Comparative Belt Tension Specifications (Kevlar)

1999 (1⅝”-wide 136-tooth USA) Rear Belt Tension Specification

2002 (1⅛”-wide 135-tooth USA) Softail (Kevlar)

2005 (1⅛”-wide 133-tooth USA) Dyna (Kevlar)

2007 to 2008 Rear Belt Tension Specifications (Carbon Fiber)

There have been changes to tensions (belt deflection), and procedures to obtain correct tensions because of the new type of belt used. The bike should be at ambient temperature with the transmission in neutral. As always, check and adjust in the belt's tightest spot. Check tension at 10 lbs pressure midpoint at the bottom of the lower belt. H-D tool (H-D-35381A) will assist the inexperienced.

## 2009 140-tooth Rear Belt Tension Specifications (Carbon Fiber)

## TC96 Primary Changes

The 2007 to present Big Twin primary system also has many changes, most of them carrying over from the 2006 Dyna. I have talked much about the new primary reduction ratios, but there are many other changes, such as the starter ring gear moving inboard on the outer hub so that there is no longer a need for a jackshaft assembly. This assembly, in various designs since its introduction on the first electric-start Harley, the 1965 Panhead 1200cc Big Twin has been around a long time. The starter solenoid kicks out the starter drive along the jackshaft to engage the ring gear. Now the 2006 to

present six-speed starters, basing on the XL Sportster starter design connect directly to the inner primary housing in order to engage the now closer starter ring gear.

There is good news for starter removal. Removal is possible without disassembly of other primary components.

There are big changes to the new thirty-four-tooth engine compensator-sprocket parts. Since 1970, the spring portion of the compensator consists of a series of fixed and ⅛”-thick diaphragm spring plates. The six-speed models have a four-spring pack for both domestic and H-D International models. The compensator spring-pack was a part unto itself that fit between other mating parts in front of the alternator adjacent to the outside of the engine casing inside the primary casing.

Alternator Changes

Now the rotor portion of the alternator combines with the compensator as one part. It has 24-multi-splines in the middle to fit over 24-mating splines on the engine output sprocket-shaft. This unit saw first introduction with the 2006 Dyna, although the rotor capacity increases for 2007. The welded-together compensator-rotor unit rotates with the engine, the rotor spinning over the fixed stator, the other half of an alternator to produce current to charge the battery.

The Softail and Dyna models upgrade with a new 40-amp alternator assembly. The rotor magnets increase to twelve magnetic units.

This stator and rotor assembly will retrofit to the 2006 Dyna models only.

In order to service as in replace the rotor, stator, or both

on the rubbermount FLHT; removal of the inner primary was not necessary. It now requires removal.

There is no longer a chain inspection cover on the outer primary, which is a drag, to check primary chain tension. This elimination of the cover results because the primary adjustment is no longer a mechanical exercise but done with an automatic mechanical primary chain tensioner. I predict problems in this area in the future although after some hard miles on my Streetglide, it is no worse for wear.

There is also a new primary-to-engine gasket and securing system bolting the inner primary to engine and transmission.

Primary and Transmission Fluid Requirements

The FLHT touring models in 2007 require 45 oz of primary

fluid if dry and 38 oz for wet capacity. Wet capacity means if draining the FLHT primary during normal servicing, the technician uses the wet fill capacity because there is residue oil left inside the casings coating the internal parts. Therefore, use the dry capacity fill when removal of the primary is necessary for major service, such as alternator replacement. I am assuming the parts are clean. Consider this situation dry. Therefore, replace the dry capacity fluid amount in this situation.

Softails and Dynas have the same fill primary requirements of 32 oz for both wet capacity and dry capacity.

Primary service intervals now require servicing every 10,000 miles (16,000km), a doubling of the 2006, and earlier interval, of 5,000 miles (8,000km).

While we are at it, the 2007 transmission fluid-servicing

interval has doubled, increasing the change interval to 20,000 miles (32,000km.) from the previous interval of 10,000 miles (16,000km).

Whilst still in this area, there is a new double-row-output bearing. However, most will refer to this more understandable language as the sixth-gear mainshaft-transmission case bearing.

I advise having a mechanic listen to your primary for adverse noise during regular service. Noise will indicate adjustment-pad-wear, chain stretching, and malfunctions. Do not confuse this with spur gear backlash noise from the transmission that amplifies through the primary cavity.

6-Speed Transmission Noise

Some mechanics feel that untoward noise is not from the transmission or primary at all. They feel there are two noises. The first is a fifth gear clatter that emanates from the transmission. Their solution is to use a heavier grade transmission fluid. This, in combination with riding at higher rpm in each gear will certainly lessen this noise.

They feel the second noise originates in the front cylinder that travels through the primary and into the transmission. This conclusion derives after eliminating other possible causes such as there is sufficient oil pressure; compensator nut torque is within specifications; compensator springs are tight; other primary components are all within specifications, and, finally, the transmission gears themselves are checked.

However, is the transmission the origin of the noise or a mitigating factor?

This is only a problem on the 2007 models as the inclusion of a rubber-mounted rear pulley on 2008 Twin Cams largely resolves the noise issue.

Switching from SYN3 Harley-Davidson synthetic oil in the transmission to semi-synthetic 80/90-hypoid gear oil will definitely help in some cases. The indicator of wear can isolate over time by observing drained fluids, for the presence of debris in the fluid whether engine, primary, or transmission oils. Always observe drained oil for untoward particulate.

## Syn 3 Oil vs. using Gear Oil in H-D Transmissions

Although it has been, my experience that 80/90-gear oil works better in a Big Twin transmission than Syn 3 oil specifically in eliminating gear whines: others have had the opposite experience. They say Syn 3 cured a whine in their bikes that occurred when using gear oil. Service managers at

dealerships have told me that H-D has advised them to replace Syn 3 with gear oil if there were noise issues like whining. While many think 80-90-gear oil is heavier than the 20W50 Syn 3, I should explain that when both test at 100 degrees Celsius; 80-weight transmission oil is roughly equivalent to 20-weight engine oil. Furthermore, the Syn 3, 50-weight oil component will test roughly the same as 90-weight gear oil.

However, there are good reasons to use gear oil since they utilize additives like sulphur. Sulphur compounds create acids in engine oil that preclude their usage. Herein lays the difference. I feel very uncomfortable using the same oil like Syn 3 for three different applications that have specific and independent needs. An engine is hot and dirty, a clutch needs cooling with lubricity for primary chain components and transmission gears chop up molecules. Additives that help one area may be detrimental in another.

I guess that the icing on the cake is that Harley-Davidson has a new recommendation to use a newly developed formulation for their primary chain case and transmissions. It is Formula +, (99851-05).

Six-Speed Gears

The design of the full-contact helical gears is quieter. They have the added advantage in that they are also stronger with more load-carrying capability than spur gears. It is true that the H-D helical-cut second through fourth gears smooth and makes the transmission quieter.

The problem, whether perceptual or real, may have its basis in the side thrust perpetrated by helical gears angled in the same direction. Side thrust may be interfering with the spur gear in fifth, amplifying its noisier backlash

characteristic.

Further strengthening the gears is an increase in gear-diameter size. Combining with these are some larger internal transmission bearings and a stronger transmission casing.

The Factory has also been very progressive in machining gears to closer tolerances to reduce backlash noise. Even distortions caused by the heat-treating process are ground smooth. Maintenance and the time between service intervals extend because of all these improvements. In my opinion, the traditional size spur gears in previous H-D transmissions are more than capable of handling the loads and power transferred through them. However, who can argue with stronger?

The helical gears eliminate the backlash noise prevalent

with traditional spur straight-cut gears in predecessor Harley transmissions.

Evidencing the noise problem, if it is one is the fact that Harley-Davidson's Ben Vandenhoeven, plus engineers and service advisers addresses these concerns in publications like the Enthusiast in the fall 2006 edition. The question becomes; are these sounds perceptually worse because they are different from traditional H-D transmission backlash noise, or is there a mechanical problem that needs correction?

The advantage for manufacturers in using spur-type gears is that they are durable and inexpensive to manufacture.

Backlash refers to the space between mating spur gear teeth allowing clatter. Clatter occurs when not under load. However, the gear teeth force together to work (under load), in unison, eliminating the clatter when they are called upon to

transfer power through the transmission to the drive pulley. The Factory has made great strides during the past two decades in terms of spur gear fitment to reduce backlash noises.

## Helical and Spur Gears

Noise reduction is the primary reason for many changes to your Harley over the years, from the elimination of rear chains in favor of enclosed, oil-bathed ones to the evolving present belt drive. Now we have the introduction of quieter transmission helical gears in the Big Twin six-speeds as well as 2006 and later Sportsters and Buell's. This new phase of noise reduction introduces with a gear design change from spur to helical in second, third, fourth and sixth direct drive 1:1 ratio mating gear sets, which are quieter than the traditional spur-gear arrangements still in use in the remaining first and fifth

gear sets.

A helical gear drive uses angled gear teeth that eliminate backlash noise since more than one tooth engages at all times. The angular sliding engagement of helical gears results in a constant meshing of gears. However, this angular engagement also causes a side thrust. The transmission helical gears will create side thrust in one direction. Herein may lay a problem because some speculate that this side thrust necessitates the need for the use of a fifth spur gear in the Cruise Drive.

Therefore, I repeat the obvious question, if helical gears are quieter than spur gears then why are there so many noise complaints from purchasers of the new 2007 Big Twins? Defenders say that these noises are not loud at all but perceptual in nature as they are indeed different. In other words, Harley riders simply are not used to them yet.

At this point, no one is saying the H-D six-speed transmissions are going to blow up because of the noise. What are the complaints? Some current ones other than the fifth gear noise are that there is sixth-gear chatter at light throttle cruise that emanates from either the transmission or the primary. Others feel that everything else about the new primary is noisy complaining there is almost a bang sound as the engine starts and the clutch makes a clack sound as you bump the friction point with the lever.

## Lugging

Lugging refers to not going fast enough in a particular gear. Lugging produces rapid wear by the engine working too hard. Symptoms include pinging, also known as spark knock, surging, and lumpy riding conditions. The engine has difficulty operating at lower rpms in a higher a gear ratio. Despite turning the throttle, the rpm climb slowly. Gaining

speed is also a slow process. As the overworked engine slowly accelerates, it will gain speed faster as it approaches its comfort zone. This is the wrong way to exit a lugging situation.

Many new riders feel that they are being gentle with their new ride by staying in higher gears at lower speeds. Quite the contrary; they damage their engines. Downshifting at least one gear always cures lugging. For example, if the rider is lugging the engine in sixth gear, the fix is to downshift into fifth. If downshifting does not fully clear up the lugging then a further downshift to fourth is in order.

Another related annoyance is that “the straight-cut fifth gear sounds bad, but most annoying is the chatter sound in sixth while not under load. The chatter goes away when loading it with either acceleration or deceleration”.

## The Clue to a Solution

There is an innocuous addition to H-D's 2008 Genuine Motor Accessories and Genuine Motor Parts catalogue. An unobtrusive entry on page 461 lists a rubber mounted rear pulley that only fits on 2007 touring model Twin Cams. The 2008 touring models use this same pulley assembly as a stock addition. Therefore, this kit is for one year only. I highly doubt any manufacturer would go to the trouble of making a kit that only fits some models of one-year production unless there is a problem.

The Factory steadfastly maintain that the fifth gear whine or clatter noises…take your pick…are consistent with the transmission design. Different types of gears juxtaposed together give a different noise than what veteran H-D riders are used to and therefore comfortable with. This is the mantra

but many riders are not buying it and neither am I. It is true that there are different noises for veteran riders to adjust to but this situation is more than that. At the very least, this noise issue conflicts with Environmental Protection Agency (EPA), noise decibel guidelines.

I have said before that I do not view the noise as problematic from a mechanical point of view. In other words, I do not foresee Cruise Drive 6-speed transmissions self-destructing any time soon. However, it offends the ears of many Harley riders and needs correction for the EPA.

IDS: Power Train Harmonic Pulse Dampener

The Isolated Drive System Sprocket Kit

All 2008 touring models equip with an Isolated Drive

System (IDS). As well IDS part number 40287-07, is now available for retrofitting 2007 touring models, except H-D International (HDI), touring models sold in Japan, with a rubbermounted rear-wheel sprocket-pulley. The kit will work on bikes with original equipment wheels and H-D accessory wheels except the 80-spoke laced wheel (42925-04), the polished aluminum solid spun disc wheel (43760-03), or the slotted 6-spoke wheel (43932-02 chrome, 43933-02 black).

The 2008 rear drive-belt sprocket-pulley compensator bowl has six rubber dampers that incorporate to form an isolated secondary drive system. The rear pulley has six stops cast in it on the inside portion facing the rear wheel compensator bowl. Thus the rear wheel pulley, marries with the compensator bowl holder via the six stationary dampening rubbers that buttress against the pulley stops when assembled together. However, it looks as if the 2-row extended race wheel bearing (part number 9285), supporting the pulley is having some warranty-wear problems. The pulley is

redesigned in 2009.

The 2009 IDS (Isolated Drive System), rubber mount rear drive pulley compensator is now part of the rear wheel hub and separate from the rear drive belt pulley. The rubbers in the compensator absorb power pulses from the engine traveling down the powertrain. Before developing the IDS, pulses were returning back up from the former rigid mount rear pulley causing the Cruise Drive spur gears to chatter. The pulley is now just a pulley with no dampening function.

Wait a minute! How on earth, can a rubber mounted rear wheel pulley correct a transmission, primary or engine noise? Sudden movement as in rapid acceleration in the rear drivetrain causes the pulley stops to compress the rubbers absorbing energy. However, the sudden movement we will be concentrating on here will be power pulses emanating from

the engine.

Here we go again. Is this not about a transmission noise? Yes it is. The noise is coming from the 6-speed transmission fifth gear and to a lesser extent the first gear.

How this noise occurs is the interesting part. I quote part of the blurb describing the IDS in the aforementioned 2008 H-D accessories catalogue, "…(The IDS) improves rider comfort and ride quality by dampening the mechanical noise and vibration caused by the engine's torque pulses…the motorcycle is quieter while accelerating, shifting and cruising." 4

I do not notice any untoward vibration that needs fixing on my 2007 FLHXI. However, vibration reduction is not always seat-of-the-pants noticeable but it does improve

quality of the ride experience. Nor do I notice any noise from acceleration inside the power band since this seems to cure the fifth gear clatter on many 2007 Twin Cams. Furthermore, I am not aware of any noise while cruising unless the motorcycle is lugging or approaches lugging conditions whilst riding in too low rpm especially in fifth gear. As for shifting,

I can feel and hear the improvement of the Harley clunk reduction with the IDS. The rubbers absorbing the sudden strain on the drivetrain softens the traditional Harley clunk when engaging a gear especially from neutral. Any noise reduction assists in meeting EPA dictates.

I think the key phrase from the H-D catalogue quote is dampening the mechanical noise. To me, the other feel-good marketing words bury this phrase in between.

Dampening mechanical noise is the reason for the IDS

design and implementation. The intention, I am sure, is to cure the fifth gear clatter in the 2007 Twin Cams. I think the real reason for the Isolated Drive System is a powertrain harmonic pulse dampener that stops returning refraction back up the powertrain thus silencing the Cruise Drive fifth gear clatter.

I feel that the noise does originate as a power pulse emanating from the engine out the sprocket shaft. The fifth spur gears in the Cruise Drive 6-speed are the vulnerable gears that rattle together from the engine power pulse.

However, I think it is not the pulse travelling from front to rear making the noise but a returning pulse from the rigid mount rear pulley back up the bike hitting the fifth gear set from

I also think the new engine compensator whose function is to dampen the engine surges has difficulty in doing so. This is why the noise is not an issue in earlier bike models. Interestingly, the 2009 compensator is now 7% stronger. Some hope, this move will control power pulses that made the IDS rubber mounted rear pulley necessary to dampen Cruise Drive gear noise. We shall see.

Helical cut transmission gears do not transfer or accentuate the refracting pulses because they are always in full contact. Therefore, they cannot rattle together. However, spur gears have necessary space between them to absorb heat expansion and cold contraction.

The new design primary acts as an amplifier to the returning refraction pulse spur-cut-gear rattle.

The clatter is the end-result and not the origin at all.

There are no Issues with the 2006 Dyna 6-speed

Why is this not a burning issue with the 2006 Dyna where the Cruise Drive 6-speed had first use?

I think a number of factors easily explain this. The Dyna is lighter, its riders more prone to higher rpm riding. We already know that riding at higher rpm in fifth gear will lessen and/or cause gear clatter to cease. Dyna riders do not smell the roses as much as the big touring bike owners do. Furthermore, the Dyna rear pulley is two-nubs bigger (68), than the 2007 Twin

Rear pulley's with 68-nubs, resulting in a higher secondary ratio of 2.13:1 for the 2006 Dyna versus 2.06:1 for the 2007 Twin Cams with a 66-nub rear pulley; a final drive gear ratio of 2.88:1 versus 2.79:1 and an overall gear ratio of 2.96

versus 2.70. Therefore, the refraction whip is pulsing with a different wave configuration.

Lastly, the Dyna and Softail primary system is a different shape and length from the touring models. The internal webbed support structure and dampening system is also of a dissimilar style. The echo chamber effect will then mute. I think the final answer of why the problem occurs primarily in only 2007 touring models is a combination of factors that come together with each one alone incapable of causing the power pulse refraction noise.

First, the secondary, final, and overall gear ratios are unique to 2007 and 2008 Twin Cams. The 2009 Twin Cams return to a 68-nub rear pulley.

However, the 2008 touring models have an IDS system installation right from the factory while the 2007 models have

the traditional rigid mount rear-pulley sprocket.

The power-side-sprocket-shaft engine roller bearing in use since 2003 has a part in transmitting power pulses. Ordinarily, the engine-compensating sprocket can cope with these power pulses in combination with the sprocket-shaft roller bearing, especially after realizing and checking the crucial torque specification procedures of the compensator nut.

In contrast, the tapered Timken sprocket shaft bearings in use up until 2003 allow up to and beyond 400 ft-lbs torque; 200 ft-lbs; or however much the backyard mechanic can muster with no ill consequences. However, the kicker that brings this noise problem all together is the supersize primary gear ratios relative to the conservative ratios in predecessor Twin Cams. H-D moves from a compensator engine sprocket of the antecedent 25-teeth up to a radical 34-tooth. This combination overwhelms the compensating sprockets capacity to absorb power pulses as effectively as before. Thus the 2009

compensator is now made 7% stronger.

Higher gearing always encourages the possibility of spark knock also, especially when overcoming the effects of the massive sprocket while traveling up the rpm range.

The pulses roll down the powertrain only to return via the rigid mount energy- transferring rear pulley sprocket particularly at lower rpm levels. The returning harmonic pulses find the absorption point of least resistance, which, in this case are the spur, cut fifth gears. The IDS cure is to rubber mount the rear pulley sprocket to absorb traveling pulses to prevent their transferring back…or, at least, disrupt the wave pattern of the pulses.

I feel that Harley-Davidson, in order to preserve traditionalism of design, goes to inordinate lengths to mask

symptoms as opposed to curing some problems. However, sometimes the cure will contravene traditionalism.

## A Possible Impending Trial on 5th Gear Noise

A frustrated 2007 Harley owner from Maryland, Russell Hall writes in a series of emails to me “In one of their correspondences to me, they (H-D) admitted that they corrected the issue in the 2008 models. However, they still refused to correct the problem in the 2007 models…A little more up-date on my case against H-D. I just received their answer to my complaint and they have requested a Jury Trial, I don’t get it, they could have fixed the bike for what that cost them...Anyway my Attorney is Mark W. Howes who is located in Annapolis, Maryland, phone # 410-266-1041. I share this with you in case you have received any other complaints or inquiries about the 5th gear whine from anyone who may live in Maryland and may be seeking some assistance. There must be more to this with H-D for them to

go to this expense to defend an issue that is an easy repair for them by installing an IDS.” 5

Power Train Absorption the Cure for Fifth Gear Clatter Origin

So, now there are compensator sprockets at the front and rear of the powertrain to curb the fifth gear clatter. Therefore, the spur cut gears are not the origin of fifth gear clatter but the end-result of harmonic travelling pulses traveling unhindered back and forth much as exhaust pulses We will see over time how reliable the rear rubber mounted pulley is although the pulley bearing seems to have some wear issues already.

Japanese bikes have been using this methodology for decades.

V-Rods spearheaded the IDS system for Harley-Davidson and so far so good. However, the high rpm V-Rod torque curve is different from the grunt Twin Cam.

The last time H-D attempted a rubber-mounted compensator was a long time ago with the Shovelhead FXS Sturgis with disastrous results.

Rubber technology has improved plus the IDS rubbers are larger and doubled in number.

So, if experiencing the gear clatter on a 2007 FLHT series Twin Cam, then I feel confident the IDS is a fix for dampening mechanical noise.

In 2009, H-D redesigns the IDS. The rubber mount rear drive pulley compensator is now part of the rear wheel hub and separate from the rear drive belt pulley. The pulley is

now just a pulley with no dampening function.

Furthermore, the engine compensator assembly that absorbs engine shock has been bottoming out. In 2008, there have been temporary fixes by adding an extra spring/shim. The 2009 compensator is now 7% stronger.

The H-D 6-speed Cruise Drive

The first Cruise Drive transmission in the 2006 Dyna was initially under a cloud of doubt for most of the year. H-D instructed customers to bring it in for accelerated service intervals while the service reps looked at a suspect bearing. It does not seem that this is evolving into something more, as an accumulation of miles ridden will yea or nay this most improved model of 2006.

I asked for and received a copy of the service addendum. Harley is monitoring the issue and has revised the service interval from 20,000 miles down to 5,000 miles. The gist of this being that the consumer pays for a service and inspection every 5000 miles, and then Harley replaces the 2006 Dyna bearing every 15,000 miles at no charge to the motorcycle owner. I have heard of no real problems as of September 2008 and suspect that this potential issue has become a non-issue.

The 2007 TC96 is overshadowing the great strides made by the Factory with the 2007 EFI Sportster. However, ithe 2007 TC96 is also under a different cloud of suspicion surrounding the six-speed transmission. It seems that the IDS rear pulley is resolving any concerns. The first and successful development of this rubber mounted rear pulley is with the V-Rod VSRC.

Cruise Drive Shift-Detent Modification

The one corrected glitch in 2006 was a recall for the Dyna transmission shift-detent arm and roller with mating spring. The detent arm and related parts are crucial to easy and positive gear shifting in the transmission. The new 2007- detent assembly (33409-06B + spring 33389-06) retrofits to post-recall 2006 Dynas and sells as a kit for the 2006 six-speeds. This new detent assembly is an attempt by the Factory to make shifting easier and give a feel-good impression with the new transmission. The transmission does feel better in my opinion relative to what went before.

Inertia Drawback with H-D 6-speed Integral Gear Clusters

However, there are the negative effects from the physics principles of inertia and momentum since there is a huge rotating mass that must turn whenever the mainshaft turns.

The one-piece first-through-fourth gears combining with the mainshaft are a great deal of weight and mass to move from a stop (inertia), and a lot to stop when already moving (momentum). If the gears were all independent of each other and the mainshaft, there would be a lot less work and expended energy. As Bert Baker says, “I believe that Harley’s 1-piece mainshaft with 1st, 2nd, 3rd, and 4th gears as an integral part is a big shortcoming with the cruise drive design. It is responsible for clunky audible shifting because it is desirable to have the inertia of the mainshaft/clutch hub as low as possible.” 6

## Changing the Gear Shift Points

The Cruise Drive six-speed moderates shift points, thus eliminating any big gaps between gears. A further bonus is a

better, more comfortable ride because of substantially lowered engine rpm. At 75 mph, sixth gear reduces rpm by a whopping 11%, from about 3225 rpm to 2850 rpm. The impact of this reduction on a high-mile day or a long trip is wonderful.

I talk elsewhere in this chapter about breaking-in my new 2007 FLHXI at speeds unheard of in Big Twin annals. Again, I say this is a tremendous bonus to the new six-speed, despite the many complaints.

Of course, the reason for this decrease in rpm in sixth gear…when the ratio is 1:1, the same as the 1:1 ratio in the fifth and final gear of the five-speeds…is the increase in primary sprocket sizes. This reduction claim in rpm is true and very comfortable in my riding experience with my 2007.

This was initially confusing to me since I have been riding

with aftermarket six-speed transmissions with overdrives in all my bikes for the past eight years. I knew the H-D six-speed in my current 2007 FLHXI was a direct drive 1:1 ratio, but it does not feel like it. It feels like an overdrive.

## 6-speed Increase in Torque Transfer Capabilities

Therefore, an advantage of the Cruise Drive is matching the bigger torque TC96 engine with modified gearing, both inside and outside the transmission.

### Wider Diameter 6-speed Gears

This occurs in tandem with an increase in 6-speed transmission gear strength. The torque capability of the Cruise Drive is almost 30% higher than the five-speed transmission. This is due to wider diameter gears.

## Higher Final Gear Ratios

The six-speed gearing results in a higher final drive ratio. The rider enjoys a seat-of-the-pants experience on a motorcycle faster than its predecessors do. The final drive ratio for 2007-2008 is 2.79:1, compared with 3.15:1 for 2006 five-speeds. Therefore, the engine turns 2.79 times for one rear wheel rotation in 2007-2008 compared with 3.15 times in 2006. In 2009, the TC96 reverts to the 2006 Dyna final drive ratio.

I would have taken issue with all this stuff before riding a 2007 Big Twin but am quite impressed now having experienced them.

## The BAKER TB6 TorqueBox 6-speed Transmission

An increase in torque capacity for new transmissions seems to be a next frontier. Introducing the strongest H-D transmission to date is BAKER Drivetrain with their TB6. The acronym TB stands for torque box, which BAKER refers to as the TorqueBox.

Bert Baker claims the TB6 rating is for 250 lbs of torque resulting from larger gears and a new case design. Compare this monster with Harley-Davidson's very strong new six-speed torque rating is 125 lbs. The six-speed's torque has a 28% increase over the five-speed. Therefore, the torque rating of the five-speed is approximately 98 ft-lbs. Torque rating increases are due to wider-diameter gears.

The H-D 5-speed and almost all 5-speed and 6-speed aftermarket transmissions have a center distance of 2.5". Center distance defines as the mainshaft-gear center distance

to the mating countershaft gear center. The H-D Cruise Drive 6-speed center distance is 2.8”. This .3” center distance enlargement explains the torque-transfer capability increase in the H-D six-speed over all previous five-speeds. The TorqueBox center distance between gears is a remarkable 3.0”, a full .5” bigger than the H-D 5-speed and .2” bigger than the Cruise Drive 6-speed.

However, this is a complicated subject, and I feel that the reader should not take these ratings as gospel but use them comparatively. They are accurate in explaining that there is an increase or decrease; the exact amount is up for discussion.

The five-speed seems puny in this ratings game, but it has proved to be very strong and durable since its inception in 1980. The 5-speed for the entire comparative faults that I will list as I continue is an excellent transmission.

I talked to Bert Baker about torque ratings. “Torque ratings are voodoo. Torque ratings are established on a transmission dynamometer. A dynamometer schedule of a particular company is usually a guarded secret…A typical schedule might have 10 hours in first, 50 in second, 100 in third, 150 in fourth, and 350 in fifth for a five-speed transmission. In each gear, the dyno will put a percentage of total intended engine torque through the tranny. In first gear, 75% of the engine torque might be used because 100% traction in first gear is rare. In fifth gear, the tranny sees 100% all the time so the dyno schedule might use full torque for the 350 hours of run time. For BAKER Drivetrain, torque ratings are established relative to other products.” 7

Bert changes gears and explains further “The most

influential factor related to bigger torque transfer capabilities is the center distance between the countershaft and the mainshaft, which in turn will dictate the diameter sizing of the gears. The H-D five-speed from 1980 to 2006 was a 2.5” center distance transmission. Center distance is the biggest characteristic to consider for a relative idea of torque capacity. All transmissions in the aftermarket base off of the 2.5” design except the TorqueBox. The BAKER TB6 is a 3.0” center distance transmission. This is a very large transmission for motorcycles. The new H-D six-speed is a 2.8” transmission, which is considerably larger than the 2.5” one that it replaces. The left-side bearings on the new H-D tranny are larger than its predecessor, and that is good. However, the right-side bearings in the door are smaller and retained by beveled snap rings. We are greatly confused by this. The door is the Achilles heel of this new transmission as it relates to performance. Smaller bearings are not good for torque capacity and beveled snap rings will pop out under severe duty with helical gears.”

The H-D 6-speed is the Strongest to Date

However, putting things in perspective, I do not think this will be an issue for most riders as the transmission is stronger than the five-speed, which served the Harley rider well for twenty-six years.

We rebuild transmissions all the time at Heavy Duty Cycles. Except a weak spring upgraded by the Factory, we find the H-D 5-speed transmissions to be durable and long lasting.

Now, would you use the new H-D six-speed at the drag strip with a big beefed-up motor? Well, the Cruise Drive is stronger than the 5-speed and most aftermarket five and six-speed transmissions.

## The TorqueBox Transmission is the Strongest

However, I cannot imagine the engine that could beat up the TorqueBox. For those that love power, the TorqueBox is a welcome addition to our world.

Of course, Bert has a fix for the Cruise Drive, which I will describe in a bit.

## Acceleration Capability Increases

The Cruise Drive transmission is all about improved roll-on acceleration, most noticeable off the line. However, I did not initially appreciate just how much better the Cruise Drive is at passing other vehicles. As I left my riding partners in the dust while passing a tractor-trailer, their enthusiastic later comments made me appreciate this fact.

# Acceleration Comparison

The figures are approximately accurate to within .5 seconds and I list them for comparative purposes only. Real-time gains are impressive. White-knuckle-passing syndrome around tractor-trailer substantially reduces.

I will say that it is a pleasure to experience this acceleration increase. My riding aggression catches me time-and-again without enough juice to roar past an 18-wheeler on previous Big Twins. White lining between tractor-trailers traveling opposite ways is not for the faint of heart. Doing this in heavy rain is real fun!

It is wonderful that the Factory for the first time is matching powerbands to the needs of the rider. I talk ad infinitum about this in performance articles and am very happy to see the Factory addressing this simple but elusive concept.

Of course, the TC96 is going to be much faster with some performance modifications that will also match air-requirement delivery into the combustion chamber with the swept volume of the cylinders in order to burn more gas. In other words, when performance mechanics put higher flowing heads, with an increase in compression ratio, performance valve springs combining with higher lift and longer duration gear driven cams, and, of course, a self-tuning, emissions-free ignition we are going to have one happening motorcycle. The result will be much higher horsepower and even higher torque with a matching transmission.

I already know that my air/fuel delivery-modified TC96 will be very much faster than those dealer-upgraded 103's that, once again, neglect air delivery as they have since the advent of the Twin Cam in 1999.

From a functional and performance perspective, the 2007 six-speeds are great and in my opinion is the best-matched transmission of all time to a Harley engine. However, the noise complaints need addressing; they are a nagging problem for a couple of reasons.

- Is the noise indicative of premature wear and impending doom? I do not buy into the doom scenario but time, one way or the other, will resolve the concern about premature wear.

- If there is no wear concern, there are still two problems: one, consumers are not happy with what they perceive as an annoying problem;

- Two, the transmission may not have met its EPA-directed objective of being quieter than the predecessor all-spur gear five-speed.

One longtime Harley owner that I cannot dismiss because I respect his mechanical abilities and Harley experiences complains that you must be going 2500 rpm for spur noise to go away in fifth gear. He feels that the noise is a result of being lulled by quiet helical gears that are full contact all the time with no gear lash noise and then shift into fifth with spur-type gears that have gear lash where a loud clatter will result in lower rpm. The good news is that as soon as the transmission is under load above 2500 rpm, the noise will many times disappear. My friend relates, “All the second, third, fourth, and sixth helical gears certainly reduce noise, but first and fifth must have had end-loading problems

which mandated straight cuts.”

## Cruise Drive Shift Clutches, Gear & Ring Dogs

S&S has a very interesting six-speed that also uses a 1:1 ratio in sixth gear. All six sets of twelve gears are helical to reduce noise.

There is a further reduction of shifting mechanism lash by the use of low-inertia shifting rings with seven dogs compared to H-D’s six-dog rings.

This is not the same as noise-producing spur-gear lash,

The H-D five-speed transmissions incorporate the shift dogs and indents into the sides of various mating gears. The gears themselves, direct by a shift drum on top. Three-shift forks slide along their respective shafts to engage and

disengage gears with each other. The 4-speed uses independent, lighter shift clutches to engage the gears. The 5-speed uses the bulky gears to engage other gears. The Factory heralded this as an efficient improvement, in the day.

With the six-speed, H-D has gone back to a shifter clutch mechanism called dog ring using the V-Rod style-design. The shift drum moves to the side from the top, using the Sportster XL/XB shifter design. There are three dog rings in the H-D six-speed, two (first/second and fifth/sixth) with dogs on both sides and one (third/fourth) that has side- pockets. The pocket dog ring has no dogs or lugs but six indents on either side. The advantage of these dog rings involves less inertia in moving them back and forth to engage with their respective gears. This compares with having to move whole gears to engage other whole gears as in the 5-speed.

This 6-speed design relates to the antiquated 4-speed system, albeit a more sophisticated version! In order to lighten the dog rings with corresponding less inertial resistance to movement, they ride on guide hubs. There are three guide hubs, one specifically for each dog ring. They are not interchangeable because of shaft/guide hub thickness and length. The guide hubs are multi-splined inside-and-out. The inside splines ride on mating transmission shaft splines, two on the countershaft and one on the mainshaft. The shift dog rings inside diameter splines mate with the splines of the outside diameter of the guide hubs. The result is a lightweight shifting mechanism that is nimble and has the least inertial resistance possible. This is a huge improvement over the bulky 5-speeds. Furthermore, it is a sophisticated refinement over the dinosaur 4-speeds.

The Harley Cruise Drive and BAKER's DD6 six-speed overdriven primaries spin the internal gears faster making it easier for the dog rings to engage when shifting. Six-speed dog

rings are much lighter than the moving gears of the 5-speed. There is less frictional inertial drag. This change to a lighter mechanism results in lighter, easier shifts.

As I relate earlier, the S&S, 6-speed has seven engagement dogs relative to the H-D six-dog system. S&S says that this lessens gearshift backlash, which results in easier low speed maneuverings especially in tight spots by minimizing lurching as the throttle is feathered. 9

There is good reason for this increase in engagement dogs. Back to Bert Baker on the H-D 6-speed, “The gaps between the Dog Clutch teeth and the corresponding openings in 5th & 6th gear are huge. When you lean into the throttle at low rpm, right in middle of the bikes torque range, the large mass of the mainshaft and the centrifugal force that it generates acts on the dog- teeth as they rattle around in the gear openings

and then crash against the gears as you twist the throttle. This vibration is then sent through the mainshaft into the primary, aka “A Big Echo Chamber” and will cause the primary chain to ‘snap’ at throttle input. The stock configuration of 1st & 5th being spur gears only adds to noise and vibration that is transferred into the primary.” 10

BAKER 7-Speed TC96 Transmission

Pictured is a BAKER DD7, Direct Drive 7-Speed transmission. It is a straight replacement of the Cruise Drive Harley-Davidson factory 6-speed improving it in addition to adding another gear. A combination of picture angle, shadowing, and a recess in the transmission trap door in The DD7 makes it appear that there are only 12-gears instead of the necessary 14-gears for a 7-speed. The DD7 has a shorter first gear for improving get-up-and-go. As a result, of the shorter first gear and following gear ratios, the bike will feel like it has enhanced engine work. The Baker DD7 transmission has the following ratios with comparative H-D Cruise Drive ratios in brackets: 3.76 (3.34), 2.75 (2.30), 2.06 (1.71), 1.55 (1.41), 1.27 (1.18), 1.10 (1.00), 7th (1.00). In transmission design, a low mainshaft weight is always desirable for smooth and quiet shifts. The Factory mainshaft is an integral one-piece design that includes first, second, third, and fourth gears. The heavy rotating weight and the resultant high inertia of the 1-piece forged mainshaft creates the audible shift clunk in the Factory 6-speed. The BAKER DD7 mainshaft is lightweight and only

includes the small first gear as an integral part to reduce weight. Therefore, the shift clunk substantially minimizes. Shift smoothness in the Factory 6-speed is a definite improvement over the predecessor 5-speed. The DD7 has another big improvement over the Factory 6-speed as it incorporates a new linear roller ball detent as part of the supplied billet top cover. This linear roller ball detent is similar to the technology in first use in the BAKER TorqueBox transmissions. The DD7 shift drum design helps the rider find neutral every time. The Factory 6-speed has straight cut first and fifth gears, which yields gear noise in these gears most noticeably in fifth. The Baker DD7 has a full set of helical gears for quiet operation in every gear. BAKER DD7 design attributes feature ease of finding neutral, smooth shifting, and includes BAKER speedo re-calibration. The DD7 Builder's Kits come ready to slide in the bike without any stock 6-speed case modifications, shimming of the gear set or adjustments.

The Helical Gear S&S 6-Speed

S&S Cycle, Inc. provides a 34-tooth transmission pulley relative to the stock 32-tooth one. The transmission design is for 2006 and earlier engine sprockets (24 or 25-tooth), combined with the 36 or 37 teeth clutch sprockets. S&S claims, “When (using the supplied 34 tooth transmission pulley) with a stock 70 tooth rear pulley, (it) yields a final drive ratio that is equivalent to that of most overdrive 6-speeds.” 11

S&S goes onto making the same point Bert Baker makes that a disadvantage of an overdrive transmission is that the top gear meshes with a number of gears for a total four gears. In 1:1 direct, drive transmissions like the BAKER DD6 and the S&S, all-helical gear 6-speed the input shaft better known as the mainshaft simply locks to the output gear. There is no further meshing of gears to create noise and frictional heat loss of transferring power.

Herein may lay another cause of noise with the H-D 6-speed since four gears are actuated in sixth, two fifth spur gears and two helical sixth gears.

S&S talks about turndown ratios. I quote, “In simplest terms, turndown ratio represents the range of adjustment to engine speed (rpm) and final drive output by a particular transmission. It is calculated by dividing the lowest internal gear ratio by the highest, so turndown ratios in transmissions with a 1:1 top gear will always be the same as the ratio of the first gear. An overdrive transmission will have a turndown ratio that is higher than the ratio of first gear because the top gear ratio is less than 1:1.” 12 This is important to consider because this “allows a touring rider to gear high for the highway and still have a first gear low enough to pull a loaded bagger from a dead stop. Performance riders

appreciate (this) because it allows them to launch hard at the drag strip…and still be able to ride in heavy traffic or at interstate speeds without difficulty.” 13

S&S also have calibration kits for electronic speedometers. Changing gear ratios and/or tire sizes require re-calibration of the speedometer if it is to read correctly. The S&S IST (intelligent spark technology), self-tuning ignition modules also have a speedometer calibration function for 2004-up models.

The noise complaints will not occur with the BAKER DD6 1:1 and the S&S all-helical 6-speed because fourth, fifth, and sixth gears pairs are helical gears of which, some portion always engages to prevent backlash noise. As I have said before, the S&S 6-speed 1:1 has all six gear pairs helical cut, perhaps making it the quietest transmission on the market.

Unfortunately, both these transmissions will not fit into the 2006 Dyna or all the 2007 Big Twins. The design is to replace 2006 and earlier 5-speeds.

## The BAKER F6F Helical Gear Conversion

However, BAKER has a cure for the noise complaints many associates with the fifth gear in stock Factory six-speed transmissions. The BAKER kit called the F6F, “features helical cut diamond ground gears, 28% wider bearings, an 1/8” thick bearing retainer plate, and a billet aluminum bearing All this combines to give the advantage of a quieter fifth gear, with no fifth gear whine and the reduction of drivetrain lash in fifth gear.

Remember that the H-D smaller end bearings with beveled snap rings may present problems with increased torque requirements.

Defenders of the Cruise Drive emphasize that the emanating noises are simply different ones than those familiar to traditional riders.

Consider that the transmission casing is stronger, gears are wider, most bearings are supersized, dog rings are lighter, and shift points tightened. Furthermore, there are two extra gears. Four sets of gears are now helical with two interspersed sets of traditional spur gears.

Normal Transmission Noise

Of course, the sounds will be different. Some say the noises generated by the six-speed is consistent with the above factors. I feel the main noise that many complain about in the H-D 6-speed has in part to do with the above comments

relating to different noises. More importantly, the spur gear backlash noise appearing, disappearing, and contrasting with the quieter helical cut gears as they come into play would be different from what Harley riders are used to.

However, the main cause of noise is engine power pulses traveling through the powertrain to the rigid-mount rear wheel pulley. The pulses then refract back up the powertrain causing the spur cut fifth-gear and to a lesser extent the first gear cluster to chatter from behind. This is not possible without necessary gear lash spacing between mating spur gear teeth. This situation cures with the introduction of the 2008 IDS rubber mounted rear pulley.

## The H-D Mainshaft Weighs Too Much

However, Harley has done something different from all the other transmissions available for their motorcycles. The H-D

countershaft is integral with C.S. fifth spur gear and sixth helical cut gear. When one rotates, all three must rotate. A manufacturing cost saving with CNC technology is probably the reason. This means that sixth gear output power also engages four gears or two sets of gears, the fifth spur gear set and the sixth helical cut set.

The sixth gear dog ring shifts directly into sixth gear, also known as the main drive gear. This action locks the sixth gear to the mainshaft. The mainshaft locks to the clutch hub in all situations. Therefore, there is a direct one to one ratio from the clutch to the pulley sprocket attaching to the final drive gear. Of course, the mating sixth countershaft gear must also rotate since it engages with the locked sixth mainshaft gear. However, this means that fifth gear must also rotate since fifth and sixth countershaft gears are integral.

The fifth mainshaft gear will rotate freely since a dog ring does not engage it with the fifth and sixth countershaft gears driven by the sixth mainshaft gear. This sounds confusing. I will say this in a different simplistic way. The noisier fifth spur gears drag along rattling away for the ride when the transmission shifts into the quieter helical sixth.

Torque Transfer Loss

I discussed this earlier with Bert Baker's comments regarding overdrive transmissions. In my opinion, the overdrive torque-transfer-loss also applies to the 1:1 ratio Cruise Drive. The one result will be a loss in torque transfer because of the frictional (heat) loss of horsepower of unnecessarily driving 4-gears instead of two. Bert Baker adds, "Yes, (this is true) because being that the H-D transmission is direct drive in 6th gear, like our DD6 there does not 'need' to be 4 meshed gears. Just 6th main and 6th counter meshed."

He goes on to say, “The H-D design drives 3-gears and has a 4th along for the ride. It does not utilize the advantages that a 1:1 drive ratio can give you. Furthermore, Bert says, “In a traditional Overdrive 6 speed the 4 meshed gears are needed to reduce the drive ratio below the 1:1 threshold to the .86:1 or .80:1 ratio depending on the transmission. Not so with Direct Drive.” 16

In continuing with this torque transfer-loss theme, the Harley-Davidson countershaft is integral with countershaft fifth and sixth gears that sixth gear output power also needs four gears with a resultant loss in torque transfer. Bert expounds on this comment, “…You are correct that since 5th is pressed onto the C.S. that it is always spinning at the speed of the C.S. You are correct in thinking that there is torque loss to drive that 5th main gear even though it is riding on needle

bearings and is not transmitting torque out the mainshaft". 17

Then Bert adds something I should have easily figured out, "There is also a measure of torque loss for the entire transmission while riding in 6th gear. This is due to the 1 piece structure of the mainshaft, with 1st-4th gears being part of the one-piece shaft. There is parasitic power loss to turn this very heavy shaft, which generates a great deal of centrifugal force. This bulky shaft design works to detract from the torque gains of utilizing a 1:1 gear configuration as well as making it shift like a tractor." 18 Stuff like this makes it a pleasure to have the ability to converse with people like Bert that are at the top of their game.

The 6-speed Rattle & the Internal Transmission Fix

Now, from a noise perspective, I feel this is the cause of

the sixth gear backlash rattle especially at lower rpm's since the fifth spur gears must also turn. In addition, of course, once put under load the noise will lessen or disappear to our ears. However, something was still nagging at my brain. The noise is more than this. I put the question to Bert; I am thinking this must be the reason for his development of the F6F fix? "The biggest motivating factor in the development of the F6F Kit was the noise of 5th gear at all rpm's, with it getting increasingly worse as the rpm's increase. The fact (is) that it sounds like a broke down supercharger whining down the highway." 19

I have already discussed the trap door bearing and Bert's fix that is contained in the F6F. Bert continues, "With all the helical gears having the teeth pointed in the same direction, all side thrust is transmitted in the same direction. This is most noticeable at low rpm, hard acceleration in and the first helical gear in the transmission. There is noticeable clutch

lever ‘walking’ at the handlebars as the torque band of the motor is delivering its payload down the mainshaft to the clutch. This excessive loading stress…will cause unhealthy and excessive wear and premature failure in the stock transmission, especially when stage kits and larger cubic inch motors are added to the equation. We purposely placed our diamond ground F6F gear pair to have the helical teeth pointed in the opposite direction to help counteract the side thrust, this becomes more important since the 5th counter gear is pressed on the countershaft and is a participant at all rpm’s and in all gears.” 20

Conclusion

I feel the much-improved Cruise Drive 6-speed transmission closely matches to the increased torque output of the TC96. I am very happy with the performance of the transmission compared to what has gone before.

However, I do have some issues with the already upgraded and superior dog rings that may need some improvement to lessen noise as they engage and the further effect they may have on sudden snapping of the primary chain. The fifth spur gear design may need rethinking both to counteract side thrust stress and to lessen noise. Stronger bearings and their fitment in the trap door may also become an issue over time. Lastly, the primary chamber will need some dampening measures to absorb noise rather amplifying it.

The bottom line is that I feel noise is an issue that needs addressing even though for some this is a perceptual component.

Please see Chapter XVI: 2009 Model Changes for updates to the compensator, IDS, touring gear ratio changes and rear pulley final drive ratios.

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The Baker F6F Released in 5/07.

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16 Ibid.

17 Ibid.

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# Chapter XI

## Performancing the TC88 and TC96 Twin Cams

### The Three Bottlenecks to Twin Cam Power

The three bottlenecks to achieving good power in the Twin Cam 88, the TC96 and their Beta Softail counterparts are insufficient flowing heads, weak timing chain support system, and emissions controls. Please see Chapter VIII: Eliminating the Performance for a detailed look at Twin Cam head flow, porting and polishing and various aftermarket high-flow heads. While the new timing chains, adjusters and oil pump introduced in the 2006 Dyna are superior to what went before there may be a new problem on the horizon with ovaling of

the rear-cam-support-plate parent material bearing via Moment Arm torsional forces. Please see Chapter II: Repairing the Achilles subsection, Moment Arm Torsional Bearing Wear for a complete examination of this potential situation.

## The Fourth Bottleneck

Emissions controls, while restrictive from 1999 to 2006, take on new meaning as air fuel ratios must meet the perfect for complete combustion Stoichiometric AFR beginning in 2006 with the Dyna EFI models and carrying over into all 2007 to present Twin Cams.

Excess heat brought on by closed loop mode, hot idle and cruising EFI air-fuel-ratios of 14.7:1 that may exceed well into the high 15:1 ratios on the shrouded rear cylinders is fast becoming the fourth bottleneck beginning in 2007.

The year 2010 will be a watershed year for many vehicle manufacturers with Harley-Davidson's nostalgic design being particularly vulnerable.

- Harley-Davidson's from 2006 to 2009 find it difficult but not impossible to meet exhaust emissions standards of no more than 12.0 grams per vehicle kilometer (g/km) of carbon monoxide (CO). One kilometer is a metric measurement that equals .621 miles or about ⅝ of a mile.

- Year 2010 emission standards, thankfully for the survival of Harley-Davidson as we know it, do not reduce this amount.

Year 2006-2009 Harley-Davidson's, have a limit of 1.4 grams per vehicle kilometer of hydrocarbons (HC), plus oxides of nitrogen (NOX).

- However, the problem is that the 2006 to 2009 limit of 1.4 g/km of hydrocarbons plus oxides of nitrogen will reduce to 0.8 g/km for 2010 and later.

The term later really means until the emissions dictates become even more rigorous. Once technologically meeting one goal for mass production then the next ever more restraining goal is set. The future will not be kind to air-cooled engines because increasing combustion chamber heat is the current way to decimate pollutants.

- The year 2020 will bring mandatory minimum mileage-per-gallon dictates. This will mean lighter vehicles. While Harley-Davidson, as a motorcycle will not have a problem meeting these 2020 35-miles-per-gallon dictates for automobiles there will be a trend to reduction in unnecessary fuel wasting weight.

The Softail frame and H-D Springer front ends come to mind as glaring examples of very heavy parts components. There will be much more on this disturbing subject for us traditionalists in Chapter XIV: The

## TC88 and TC96 Similarities and Differences

Much of the performance principles and ideas I will apply to the 1999-2006 Twin Cams will apply also to the TC96. The TC96 differs from the TC88, certainly by eight cubic inches that do not provide much of a horsepower increase but excels with torque amplification. There are three other major changes from the TC88 to the TC96, which are true closed loop EFI and dramatic gear ratio changes combining with a 6-speed transmission.

## Closed Loop EFI

The first is the introduction of closed loop EFI for cruising and hot idle modes in the TC96 and CVO 110 in 2007. True closed loop EFI actually has first use, in the largely successful guinea pig 2006 Dyna TC88's. Closed loop EFI uses a Stoichiometric 14.7:1 air-fuel-ratio (AFR). For gasoline, this AFR accomplishes most complete combustion.

This sounds good at first blanch since the more air/gas that burns, the more power creates. However, Stoichiometric AFR is blistering hot; too hot for an air-cooled in-line cylinder engine where the shrouded rear cylinder must endure as high as a 16.0:1 AFR. The TC88 is like the good old days, there is much more freedom with emissions controls as restrictive as they seemed at the time. I address Stoichiometric AFR closed loop EFI in Chapter III: H-D 110 Cubic Inch subsection, EPA Perfection: Stoichiometric Air Fuel Mixture and Chapter IX:

Fuel subsection, Electronic Fuel Injection and elsewhere in this Volume II: Performancing the Twin

## Supercharged Gear Ratios

The 2006 Dynas and 2007 and later Twin Cams have a previously unbelievable 34-tooth engine sprockets and a 46-tooth clutch hub sprocket giving a primary ratio of 1.35 engine rotations for 1 rotation of the clutch hub and therefore 1 rotation of the attached transmission mainshaft that the hub sits on.

Modern clutch hubs (1990-present) have spline centers that mate with a spline outer diameter of the transmission mainshaft that extends out from the transmission and into the inside of the inner primary casing connecting the primary powertrain to the transmission powertrain. Evo models from

1994 to 1998 and TC88 models from 1999 to 2006 (except 2006 Dynas), have what we once thought was normal 25-tooth engine sprockets combining with a 36-tooth clutch sprocket giving a primary ratio of 1.44 engine rotations for 1 rotation of the clutch hub and the transmission mainshaft. Previous Evolution Big Twins from 1985 to 1993 had a 24-tooth engine sprocket combined with a 37-tooth clutch hub sprocket giving a primary ratio of 1.54 engine rotations for one rotation of the clutch hub and the transmission mainshaft. When performancing these 1994 and later models at Heavy Duty we routinely retrofit the 25-tooth engine sprocket to a 24-tooth sprocket for better off the line get up and go.

Sprocket/pulley Sizes

Primary: Secondary and Final Drive Ratios

The reason that 2006 Dynas and all 2007-2008 Twin Cams have differing gear ratios is because of a 2-tooth/nub reductions in the 2007 and 2008 rear wheel pulley. The 2009 Twin Cams return to the 2006 Dyna gear ratios. The 2009 Twin Cams use a 68-nub rear pulley.

I cover the supercharged gear ratios found in the 2006 TC88 Dyna and all 2007 and later Twin Cam TC96 models at length in Volume I: The Twin Chapter VII: The Controversial Harley 6-speed Transmission. The subsections to read are HP and Torque Mismanagement; Primary Gear Reduction Ratio; Secondary Drive Gear Ratios; Final Drive Gear Ratios and Internal Transmission

The 6-speed Transmission

I believe Harley-Davidson developed the 6-speed transmission in response to the aftermarket versions available

since circa 1999 with the CCI offshore Rev Tech (www.customchrome.com) offerings quickly followed by BAKER Drivetrain's (www.bakerdrivetrain.com), all-American made 6-speeds in a variety of gear ratio configurations and strengths. Then others came like Midwest Motorcycle Supply (www.midwest-mc.com), offshore inexpensive versions driving prices down dramatically followed by the Screamin' Eagle (www.harley-davidson.com), versions, which I think are manufactured offshore. As a passing note, at the time of writing Custom Chrome is in bankruptcy protection but there are offers to purchase the Rev Tech line of performance parts on the table. I mention this because the web address may become defunct or change.

The TC96 Power Riddle

The 2007 Big Twin powertrain offers a conundrum.

Radical gear ratio changes are the big reason for the increased performance seat of the pants feel of these new models in conjunction with more torque.

However the riddle is stated in two parts;

1. The bigger engine does not offer any more horsepower.

2. The new 6-speed transmission does not come with an overdrive gear.

Rear wheel horsepower does not appreciably increase! However the combination of a torque increase and matching the transmission gear ratios to the new primary drive ratios make the bike very much faster than predecessor Big Twins. I cannot compare and discuss the new 2007 TC96 engine in isolation but must do so within the context of the new primary sprocket ratios and the 1:1 ratio 6-speed transmission

also. Acceleration improves with the TC96 because the increase in displacement improves torque that capitalizes on the optimization of the marriage in external and internal transmission gear ratios.

Engine rpm speed reduces in top gear at 75 mph, by slightly over 10 percent, from about 3225 rpm to about 2860 rpm. Lowering rpm will result in a less tiresome and enjoyable ride. Of course, the reason for this rpm decrease in gear when the ratio is 1:1 the same as the 1:1 ratio in the fifth and final gear of the 5-speeds is the extraordinary increase in the sizes of the primary sprockets.

Performance builders are attributing what they feel is a loss in horsepower in the new engine to this increased size compensator sprocket. However, there is an appreciable increase in power but in different areas than earlier models.

Roll-on acceleration during normal highway passing speeds of sixty-to-eighty miles-per-hour significantly increases. If comparing a 2007 TC96 Bagger with its 2006 TC88 counterpart in it will only take five seconds to go from 60 mph to 80 mph relative to the TC88 needing around seven seconds to do the same. The figures are approximate but close depending on state-of-tune and other variables.

In a roll-on, the comparison is more dramatic with about 7.15 seconds for the TC96 and about 10.8 seconds for the TC88 Bagger

Head Flow

The TC96 has marginal improvement with cylinder head flow over the anemic TC88 flow.

It feels that I have said the following a million times.

The smaller 1340cc, eighty cubic inch Evolution models flow much more air than the bigger 1450cc, eighty-eight cubic inch Twin Cams.

The Evo D-shaped combustion chamber head is the first to use a squish band in combination with a flat-topped piston to squeeze the air-fuel mixture turbulently into a smaller combustion chamber.

Turbulence increases the Percentage of Burn

Turbulence mixes the air and gas into more correct proportions than the predecessor inefficient Shovelhead pure hemi-head design. A higher percentage of fuel burns creating the heat and thus pressure via heat expansion to drive the piston down on its power stroke to create power at the rear

wheel. The power at the rear wheel where the tire contact patch meets the asphalt is more correctly torque.

Perhaps over simplistically, torque gets the bike moving and horsepower keeps it moving.

Two Methods to Power Gain: Cubic Inches and Air Flow

The first is the Go Big or Go Home crew who believe more cubic inches is the way to go. They are correct in their assessment except that it is not the only way to go. Increasing airflow is the other way to gain much power. Combining cubic inches with more airflow really makes an engine cook!

Cubic Inches

Cubic inch increases create more swept volume in the cylinders by increasing the piston stroke by relocating the

crank pin closer to the perimeter rims of the flywheels. Therefore, the pistons push up further in the compression, exhaust strokes, and travel down further during the intake and power strokes. The rotating flywheels pull down the intake piston stroke but the power stroke drives down via expanding heat.

The TC96 bore size does not change over that of the TC88, as I will soon discuss. Nevertheless, we can refer to the TC96 as a stroker since it increases in stroke or piston travel from the TC88 stroke of 4.000" to 4.375" (4 ⅜"). However, the piston does not travel .375" extra both at TDC (top dead center), and BDC (bottom dead center). This would mean a stroke increase of .750" (¾"). The stroke increases half the .375" or .1625" at TDC and the same at BDC to arrive at a .375" travel increase.

Like bore, which is the measurement of diameter, the increase is half on either side better known as the radius to make the whole. A 3 ¾” bore increasing to 3 ⅞” or ⅛” has a diameter or bore increase of this amount but a radius increase of half the amount or

Increasing bore size or diameter of the cylinders is the other way to increase swept volume. The very popular upgrades from the TC88 to 95 cubic inches and the TC96 to 103 cubic inches, an increase in 7-cubes for both engines by simply boring the stock cylinders from a diameter of 3 ¾” to 3 ⅞” (⅛” or .125”), always seem to neglect the necessary head flow increase that must now accommodate an even bigger swept volume.

The reader now discerns that both the TC88 and the TC96 have the same cylinder bore of 3 ¾”. In fact, the cylinders are the same. It is a pretty-smooth move on the part of the Motor Company; talk about efficiencies and economies of scale.

Hence the primary reason, we the consumer did not get a 95 cubic inch engine in 1999 and a 103 cubic inch engine in 2007.

The marketing genius of Harley-Davidson offers the all-time most popular upgrade to the riding public: making oodles of money from the up selling and in the same breath effectively cutting out the aftermarket. I realize none of this has much to do with our performance-minded objectives but once again, I will say that leaving an increase in head airflow out of the equation is wasteful and counterproductive. There is not much point increasing the swept volume of the lungs capacity for air unless head flow breathing increases dramatically. We are left with a faster motorcycle but not as fast as it should be!

Please see the next section, Chapter XII: Cubic Inches:

Strokers and Big Bore for lengthy discussions on more power through increasing engine displacement.

## Swept Volume

The piston sweeps up the cylinder on the compression and exhaust strokes and sweeps down an equal distance on the intake and power strokes that combine for of crankshaft rotation to complete the four strokes in one engine cycle. This area that can measure from, take your pick, the piston top at TDC to the piston top at BDC of travel or from the bottom of the lower oil ring at TDC to the top of the upper compression ring at BDC. For me, the ring method is better for hemi-head or performance piston-dome designs while if the piston is flat-topped without valve pockets either method works well. We are simply measuring the cubic inch or the metric cubic centimeter volume of the cylinder. Obviously, the volume increases with either stroking or big bore or commonly with a combination of both.

# Air equals Power

The combustion chamber lung needs to inhale more air through the induction system and intake port as well as expel waste out the exhaust port and exhaust system. How well this process occurs determines power. How can more fuel introduce and burn unless it is able to combine in correct proportions with the oxygen in air?

Therefore, air equals power. This is why anemically flowing heads are such a restriction to power. Head flow is central to making torque and horsepower!

Head flow depends on many components inside the engine to increase airflow on top of excellent port design and valve sizing to allow sufficient volume through to service the needs

of the engine. Assisting head flow are the cams with their four individual valve trains servicing each valve. Performance cams open the valves further (lift) and longer (duration). The cam lobes dictate how much the intake and exhaust valves overlap each other in their functions in order to take advantage of physical principles like inertial drag and momentum to stack up the air/fuel beyond 100% cylinder fill.

Feeding the ports depends on an air filter that will flow much air. Secondly, we need an internal air cleaner design that changes the relatively slow moving turbulent air into laminar flow feeding into the induction throat to increase velocity and volume.

The naturally aspirated induction that depends on the physical principle of vacuum whether carbureted of fuel injected but it needs to channel enough air and fuel to satisfy the combustion chamber via the intake head port and the larger intake valve.

## The Lowly Exhaust System Does so Much

The exhaust must be able to handle the volume forced out of the smaller exhaust valve and its subsequent port. The exhaust does so much more. The exhaust times back pressure Kinetic pulse waves that travel back up the exhaust many times to bounce off a closed exhaust valve for fear of bastardizing the fresh intake charge. Exhaust backpressure also has much effect on torque and positioning of the powerband.

The last component support part for flowing heads to do their job is the ignition that must time spark in all sorts of conditions to burn the fuel at the correct time. This will create maximum heat, which expands as pressure to drive the piston down on the power stroke to send the torque and horsepower

down the powertrain to the rear wheel.

A stock engine modified with head porting and exhaust port polishing with the correct and coordinated support components can create much power without increasing cubic inch displacement. I have beaten many a big bore-stroker with headwork, cams, induction, exhaust, and ignition. Please visit Chapter VIII: Eliminating the Performance Obstruction and Chapter IX: Fuel subsection, Air Cleaner plus Chapter XIII: Forced Induction for encyclopedic information on air delivery.

Power Delivery Tuning

Properly tuning the motorcycle is the forgotten ingredient to power by many.

Whether performance derives from cubic inches, airflow or

usually both, it is so important for the fast moving machine;

1. To have the support of a properly adjusted clutch mechanism that grabs and transfers the highest percentage of power back to the rear wheel.

2. Furthermore, wheels, belts, chains, and chassis must align according to specifications. Power must move straight back and not comprise with components directionally misaligned.

3. Tire inflation is crucial.

4. In addition, all those bearings must be in good shape, like wheel bearings and races, steering head bearings and races as well as rear fork bearings, races, bushings, or rubber mountings.

5. Moreover, bearing, bushing, and/or rubber fitment must be under correct preloads. Over tightening a bearing hold down fastener, bolt or nut is just as bad as too loose fitment.

6. The bike chassis and all its adjacent parts must torque to specifications to produce rigidity.

7. Cracked or broken bracketry that cannot cope with stock power certainly will further compromise increases in power.

Handling and Braking are a Necessary Part of the Performance Package

Shocks and front forks must be stiff to challenge weakening or performance robbing compression and extensions…and yet not too stiff to function correctly.

- Upgrading rear shocks and the front dampening suspension system will again assist in total motorcycle rigidity so that it will go exactly where pointed.

- Those handlebar rubber mounts should replace with stiffer and more functional polyurethane ones.

- Foot pegs must be firm and secure with slip-free rubber. Trippy billet pegs may cause a foot to slide off when least expected and sometimes-in dangerous situations.

Performance riders like a set of foot pegs in the center of the bike for control in adverse handling S-bends or the like. A wheelie most easily controls with the feet in a central position rather than splayed forward. Handling and braking must go hand-in-hand with power increases for safety and the ability of the rider to control the performance.

It is now time for another one of Donny's Blasts from the Past.

Donny's Blasts from the The Race: Wife-3 Knew "it was on"

My buddy was cock-a-doodle-dooing about he had the fastest bike. However, there is a lot more to a fast machine than engine horsepower and torque.

The first sneaky speed secret is obvious. However, all too many riders and mechanics ignore it even though it is the least expensive pony's money can buy. Rule 1: keep the powertrain in perfect tune and adjustment.

The three most important factors in winning a street race

are rider skill, rider reaction time, and weight-to-power ratio. A smaller rider on a lighter bike has a huge advantage. The fourth is a tuned powertrain.

Many will disagree with my next statement that cubic inches and other performance modifications can come in last. With professionals, power moves up the list in importance as other factors like reaction times equalize out.

At the time, my well broken in stock FXR was fast for a stock Evo. The FXR engine was loose with low frictional horsepower loss inside the engine. Compression was still very good. Any thinking rider realizes that the superior FXR chassis transfers back more torque than other H-D models. I kept the bike in proper and correct adjustment everywhere required.

I weigh 220 lbs. but with wife 3-on the back, we topped 330 lbs, which is a disadvantage in terms of weight-to-power

ratio.

Fortunately, my friend Hunk Handsome weighed 300 lbs. Hunk took his Softail and spent oodles of money on performance at the dealership. He felt that his engine had at least 50% more horsepower than mine. I knew his engine was still tight with frictional loss of power at its greatest. The Softail has an inefficient chassis design that loses lots of power going back to the rear wheel. I looked at his tires and could tell that they were chronically out of proper inflation with too much air by the wear patterns.

If I wanted to I would have let a little air out of my properly inflated rear tire, which is an old drag racing trick to give the tire more bite with the pavement. However, there was no time for adjustments with this race driven by testosterone spontaneity.

The FXR has center mount pegs that allow better control for the rider while the Softail has them out front displacing weight awkwardly as well as limiting rider control.

Riding side by side, we came to the proverbial stoplight. With nothing said, we instinctively got ready. Wife-3, held on as she felt me shift my weight to make the tire bite the pavement better. She knew it was I leaned my upper body forward also to keep the front wheel on the ground. All of this is difficult to do on the Softail forward controls.

The stock FXR took him out of the hole but wife-3 could not hold on. I felt her slipping back off the seat and onto the rear fender. I instantly hammered the front brake without lowering the rpms, throwing her forward against my back. Her grip around my waist intensified. If she knows one thing in life, she knows that I do not give up.

She now also knows that she wants a seat rail and pad.

I threw my weight forward to keep the front wheel on the road. Just as fast, right before the top of the powerband, I speed-shifted the FXR into The front wheel momentarily left the ground. I flew into third winding her out…the bike not the wife. With each gearshift, the FXR gained speed and took the hopped-up Softail clean right up to the ton (100mph).

To this day, Hunk thinks I was lying about my engine being stock, still wanting to know what work I had done.

Aggressive Riding

The problem, other than my personality trait I have with

my aggressive riding is tire grip especially when hard braking. Harley-Davidson's and their tires, maybe because of their motorcycle weight just love to break contact and slide when the wheels lock up with heavy braking. Fortunately, experienced riders can do the slide straight on. Perhaps the introduction of ABS braking on some new models will resolve this problem. Intellectually I know it will. Emotionally, it is difficult to relinquish control.

Of course, the real problem is that I am going too fast for conditions around me. Knowing when to slow down is probably the best remedy of all. As I will demonstrate, I do not follow my own advice sometimes to my peril and other times my benefit.

I feel another Donny's Blast from the Past coming on.

Donny's Blasts from the Past: Aggressive Riding in Africa

One great friend and a pretend-friend-traitor (see acknowledgements in future volumes) and I rode a circuitous route around South Africa from Johannesburg down to Durban on the ocean, up the coast, back inland through the Zulu kingdom of Swaziland up to God's Window near Kruger National Park and back down to Jo'burg. We were all riding Baggers, which were too big for where we kept winding up.

South Africa has a large motorcycling community and culture. You will not be alone riding this mystical land with biking roads for every taste from touring to dirt.

Everything you have heard about South Africa, in the Dark Continent is true; half-true or has a grain of truth. Turquoise oceans, white sand beaches, the dry savannah where big game roams, the mountainous terrain with lush foliage, a big blue

Montana like sky and the inland waters all untainted by pollution, grace the eye with a beauty and excitement that few countries afford. Of course, when on the road in the countryside, alone as a tourist, there are ten different ways to die each day also. Insects, parasites, ever-present crocodiles, territorial hippos, snakes, big cats and on it goes with the water creatures even worse. This is also home to the Great White and Bull sharks. Did I leave out human brutality that sometimes defies description?

Yet, it is a paradise; the friendliest place I have ever been with wonderful food and an intriguing history. The South African driving system bases on Great Britain's. It is opposite to North America and the rest of Europe. Riding on the opposite side of the road with jetlag is a confusing and dangerous situation. Aggression will only lead to an accident.

Road signage is professional and directive with many novel rural ones you will immediately understand such as a paw print to indicate that leopards are in the immediate area. Other signs include; Watch for Hippos, Beware of Crocs and a sign similar to North America's deer crossing sign except it is for Impalas and the myriad of other species of antelope.

A local biking group was having a ride in and around the outskirts of Jo'Burg and invited us along, which is typical of the friendliness we continuously encountered. Speed limits are similar to home but are not enforced. The riders and drivers ride according to road conditions and traffic flows. Highway and expressway speeds easily approach and go beyond 80mph in the fast lane.

One biker approached us and offered to tone the speeds down to about 80mph. We foolishly said not to slow down for us. He laughed and said, "Well, then ride your bike like you stole it!" Now my aggressive riding was the only trait that

would save me as I wound up riding in a pack at 100mph, on the wrong side of the road in a place I had never been.

Next day, I was now ready for the 375-mile ride over the low-lying Drakensberg Mountains southeast to Durban, the major city resting on the shores of the exotic Indian Ocean. We met some other local bikers that had spent the weekend in Jo'Burg and were heading home to Durban and they invited us along. We rode higher up into the arid mountains through winding roads and made another stop off the highway at an old restaurant way station. Gracing the dirt streets were native Zulu women in colorful sarongs, effortlessly walking in a slow rhythm, balancing the night's bundled firewood unassisted on their heads. The Zulu battlefields are in the area where the English and the Dutch with guns and rifles found this indomitable warrior people easier to coexist with than to fight their spears.

As we rode on, we admired herds of dozens of species of antelope roaming the Savannah near the road. We almost hit a group of wild goats grazing near the center of the highway. I soon realized that the roadways were safer for these animals since the tall grass hid the predator big cats.

The descent from the mountain pass another 200 miles from our destination was winding and became a fun and invigorating ride at 80 mph on a highway designed for speed. I never ride in rural areas at night at home because of the danger of hitting deer. The last thing I would ever think of was to do this in Africa of all places where the situation of crossing animals magnifies many times.

However, here we were as the black and I mean black African night descended rapidly at 6pm with still 90 minutes of riding left. The corners now became dangerous at the

speeds we were traveling but the local bikers moved on relentlessly. Of course, if we had have read a map or even brought one, we could have slowed down and gone at our own pace. We had no choice. What was worse, risking an accident or being potentially lost in the pitch-dark African outlands?

We sped up.

We rode into Durban passing through bad neighborhoods with no lighting except fires burning in barrels and faint figures moving towards us out of the shadows.

In unison, we instinctively accelerated. Aggression and speed became my friends.

After enjoying the beach the next morning, we rode north

on along the coast planning to stay in an oceanfront quaint bed and breakfast. I unknowingly turned off the highway into Richards Bay through an assisted area where the housing consisted of rusted sheet metal sheds as small as big closets. We unconsciously knew not to stop with the people emerging from their shacks looking as if Christmas dinner was arriving.

We soon found ourselves on a deserted dirt road going nowhere we wanted to go, as our gas tanks went on reserve. I knew we could not go back as we only gotten through by surprise. They would be ready for us. As the saying goes, “They will rob you at night and murder you in the morn.” I always wonder what happens in between.

Then the road ended!

Frustrated and very alone, we backtracked to the only

logging road before the rural ghetto that would hopefully lead to the highway. The sun was setting in the west and we rode towards its blinding rays since I thought that was where the north-south highway was. I began to ask myself one more time how I always seem to wind up in these predicaments, out of gas on a dirt road in the middle of nowhere in Africa with night setting in.

Looking for adventure and whatever came our way began to take on new meaning. I began to worry about cats in the middle of the night but later found that this area infests with a variety of very deadly poisonous cold-blooded snakes that look for a warm body. We stumbled onto the highway after dark and after some searching, found a fortified compound to spend the night.

The next morning, while lazily riding along in Swaziland, I began to look closely into the roadside rocky brush. Some fierce warthogs with their oversized tusks came into view

trotting parallel to my bike. A proud but disinterested male lion lie amongst some rocks and grass close to the roadway. Then of course, the monkeys darted amongst the trees while more warthogs ran from the noise of our exhausts.

Next day in attempting to exit Swaziland from its northwesterly mountainous region, we wind up on the most dangerous road I have ever ridden. It followed the crest of a mountain with certain death on both sides. The 15-mile treacherous dirt cambers down from the center, drawing the bike down to soft red earth flirting with the deep valley below. Riding this road takes skill and confidence. It also takes aggressiveness since speed is the only way to stay in the middle.

We finally hit the Swazi border in the middle of nowhere where the guards were in disbelief to see the big bikes ridden

by the dumb white guys. This border closes at 4pm since there is only two hours daylight left and reopens at sunrise; the roadway is much too dangerous to transverse in the dark for the normal 4-wheel trucks.

Then there is another 20 miles of dirt road along a mountainous ridge that we nicknamed Nature's Roller Coaster, where riding mistakes are not a good idea. My aggressive riding constantly gets me into trouble but in Africa, judicious aggression saved my bacon every day

The rest of the trip was exciting and wondrous but uneventful in terms of danger. I will elaborate on the whole riding trip in South Africa in detail in Volume Living the Dream.

Screamin' Eagle Engines

# Screamin’ Eagle 103 Engine

In my humble opinion, the Screamin’ Eagle (S.E.) TC103’s were a waste of money, particularly the earlier ones.

I am very critical of the S.E. CVO 110’s that began production in 2007 but they are very much better than the TC103’s from a performance viewpoint. The CVO 110’s have a problem new to Harley-Davidson that the TC103’s did not have to cope with. Leaner air fuel ratios in EFI closed loop mode creates excess heat and can cause subsequent parts damage.

The Screamin' Eagle 103 cubic inch models were the hot ticket for those who could afford them in the early 2000's. They had the same bore and stroke specifications of a TC96 to TC103 big bore conversion, a 4 ⅜" stroke, ⅜" or .375" up from the stock TC88 4.000" and a bored out stock TC88/TC96 3 ¾" cylinder bore, ⅛" or .125" to a 3 ⅞" diameter.

The TC96 conversion to a TC103 is superior to the older Screamin' Eagle 103. Head design alone makes it so. The TC96 heads while not flowing sufficient air; flow more than their TC88 counterparts do.

I have repaired many of the S.E. 103's and brought them much closer to their true performance capabilities. Unbelievably, the first S.E. 103 engines use close-to-stock fuel delivery and ignition with stock TC88 head fitment! The Factory threw on heads that flow less air than those for the 80 cubic inch Evolution onto a 103 cubic inch engine. Many

other S.E. 103 components were stock Twin Cam 88 ones like fixed length pushrods and rubbing rocker arms that, in my mind should have been Screamin' Eagle adjustable pushrods and S.E. roller rocker arms.

Cams; I address at length both in Volume I and this one, Volume II about the weak timing chest components and their frail spring-loaded chain adjusters coping with performance valve springs a prerequisite for higher lift performance cams. Of course, I have seen early S.E. TC103's with stock valve springs.

Later engines mercifully began to incorporate the Screamin' Eagle hemi-heads with performance higher seat-pressure valve springs. To me, I have never understood why the Motor Company did not use the higher flowing Screamin' Eagle HTCC heads and matching pistons but this is small

potatoes compared to other transgressions to a performance engine.

I had better clarify information about hemi-heads, as regular readers will know that I am not exactly infatuated with them.

## Hemi-head Performance

I constantly deride the hemi-head combustion chamber and it's mating domed piston as a design of inefficiency. The old-style hemis like the Shovel, Pan and Knuckle had high round domed pistons and equally deep round pocketed combustion chambers. There is no incentive for mixing air and gas in combinations that can burn. The dome interferes with the sparked flame travel over and around the intrusive piston top design.

Let us not forget the 1957-1985 Ironhead XL Sportster hemi-head, which saw slight improvement in design as the sport hotrod Harley. The combustion chamber shape was the same as the Big Twin counterparts. However, the compression ratio increased assisting in burn and power as well as unwanted detonation. More importantly, the piston dome was not rounding although it still extended far into its mating chamber. The mini-flattop one-dimensional pyramid shape dome faces the valves, while inefficient by modern standards was far better than the Big Twin variety.

Dual plugging each cylinder head assists mightily in flame travel and a higher percentage of burn in these older hemis. The older the head design, the more important dual plugging becomes; the newer the design, other factors most notably the flat top piston meeting a squish band make dual plugging less of a necessity for increasing percentage of burn.

The modern performance hemi-head may look the same as the dinosaur technology but has subtle differences that make all the difference. First, there is a squish band incorporated into the beginning of the piston dome as it graduates away from the piston crown. People may not notice it because it angles, running the circumference of the dome. There is a similar mating circumferential mating area at the combustion chamber as it extends into the head away from the head-gasket-surface area. The piston dome is not as intrusive as the hemi-piston of old as the dome is lower and not as pointed. As well, the combustion chamber is not as deep.

This modern hemi-head design incorporated into one of the earlier Twin Cam Screamin' Eagle head selections allows flame travel mostly unimpeded travel across the chamber after squishing the air/gas semi-turbulently into the chamber allowing for a better mix and thus a superior burn. This is a huge improvement over the 1936 to 1984 hemi-heads but still

not as good as, say the Screamin' Eagle HTCC heads and piston top crown design. However, where utilizing big valves and valve shrouding is a concern then this squish hemi-head becomes the choice.

We will typically see this design on the monster cubic inch engines where huge valves employ to deliver in the air and fuel and then expel the waste gases.

The Screamin' Eagle CVO TC110

The Custom Vehicle Operation (CVO) TC110 shares the TC96 4 ⅜" stroke crank assembly. The CVO 110 achieves the extra cubic inch displacement by increasing the bore from a TC96 3 ¾" a full .250" to 4.000".

Performancing the CVO means installing some parts that

should have been there in the first place. The stock TC88/ TC96 pushrods found in the CVO need replacement with adjustable performance pushrods. More importantly, roller rocker arms need installation to replace the stock rubbing arm rocker arms used in all Evos and Twin Cams since 1984. The rubbing rocker arms rub across the top of the valve stem, pushing them down to open the valves. This causes much side loading and friction pushing the valve against its housing valve guide causing premature wear and frictional loss of horsepower.

Then H-D goes the other way with valve springs in the CVO 110 that have too much seat pressure. The very good quality triple valve springs are overkill for this engine. Their design is for servicing up to .650” lift cams and yet the CVO uses .550” cams. Valve-spring-seat-pressure of the valve to its mating seat must balance with the lift of the cam. If there is too much seat pressure, the valve will slam into its seat leading to premature wear.

The CVO .550” cams are emissions ones so they have a purpose more important than creating power, which is to assist in controlling emissions. This is not the Motor Company’s fault as the EPA ties their hands in this area.

However, we need cams with higher lift, longer duration and more valve overlap to meet the breathing needs of a 110 cubic inch displacement and to get this baby cooking.

The CVO 110 uses the same 46mm throttle body and injectors as its much smaller displacement counterpart the TC96 with a full 14 cubic inch differential. The 2007 CVO 110-throttle body is 27708-06 and the injectors are 27709-06A. The 2007 TC96 throttle body is 27708-06 and the injectors are 27709-06A. The 2008 CVO 110-part number changes to 27605-08 because of the servomotor that is

integral with the throttle body for the wire-by-throttle control.

The 2008 injectors are part number 27609-01B. This indicates a change from the 2001 to 2005 injectors (27609-01A) for the even smaller displacement TC88 Twin Cams.

I think we are going backwards again. In any case, these injectors and throttle bodies are inadequate for the needs of the CVO 110.

In addition, we need more gas to assist in cooling down the stock 14.7:1 Stoichiometric air-fuel-ratios found in cruise and hot idle closed loop EFI modes. Gas cools combustion temperatures by lowering the AFR ratio.

I recommend going to the Screamin' Eagle 50mm throttle

body and paired injectors to deliver more air and fuel into the combustion chambers. The 50mm throttle body part number is 27623-05 and the injectors are 27654-06. These will only work on the 2007 CVO but not on the fly-by-wire 2008 models that require a servomotor to activate the throttle plate.

The 58mm throttle body (27623-05), and the matching injectors (29915-08), are now available in the Screamin' Eagle 2008 catalogue. The Factory recommends the 58mm body and injectors for Harley's with over 100 rear wheel horsepower. I feel the CVO 110 needs more performance work to be above this level into a good comfort zone so the 58mm body and injectors do not overwhelm the engine.

The big problem with the CVO 110 is heat. The CVO 110 needs the head alloy composition to change to resist warpage particularly on the rear head. A complete cylinder redesign is in order to equalize tie down pressures to assist in head gasket

integrity. Anything we can do to lower cylinder head temperatures will assist both in power and in engine longevity.

I also recommend the addition the Harley-Davidson cylinder head fan (part # 91550-00C, chrome; 91531-00D, black) and a one-quart larger oil tank, both to combat excessive heat as well as eliminating the too-restrictive, in my opinion, Premium Harley-Davidson 8-core oil cooler and substituting two 10-core Jagg coolers. There is nothing wrong with the H-D Premium oil cooler when used alone. In fact it is excellent but doubling the H-D oil coolers up will result in too much oil pressure

This has been a brief discussion of performancing the CVO 110. Much more information is in Chapter III: H-D 110 Cubic Inch

# Performance Basics

Much of what I will write in this section will be applicable to both the TC88 and the TC96.

Some exceptions will revolve around performance gearing. Performancing the TC96 allows the engine to grow into its supercharged gear ratios making the bike so fast that there is no place in Canada and only a few areas of the USA where it is legal to have these bikes in sixth gear at a proper rpm without breaking speeding laws. These are the first Big Twin Harley-Davidson's that can handle cruising the medium speed center lanes of the European Autobahn and use the outer fast lane for passing when clear of cars traveling upwards of 150 miles per hour.

## Headwork

While TC96 heads flow better than their TC88 counterparts do, they are sill deficient from a performance perspective.

My porting and polishing commentary will apply to both sets of heads.

Of course, given a choice I would install TC96 heads on a TC88 for a little more oomph. However, if going to this trouble, either headwork to the TC96 heads or replacement with superior aftermarket ones would be in order.

Another area is the timing chest components in the TC88 and those in the TC96.

The TC88 timing chains, spring-loaded adjusters, and sprockets need replacing with a gear drive for three major reasons.

1. The first is to replace deficient parts like the adjuster pads that may deteriorate, crack, or break at any given time.

2. The second is to correct a cam timing variance of up to four or five degrees because of sloppy sprocket to chain fitment.

3. The third is that performance valve springs overwhelm the spring-loaded adjusters and further throw off cam timing by an additional four degrees.

The TC96 is undisputedly a better system. However, cam-

timing variances still exist. Furthermore, the Moment Arm Force exacted on the rear cam outboard journal in particular by a too close sprocket and pulling timing chain may oval the parent material hole-bearings in the new cam support plate.

We want to minimize Moment Arm Force and cam timing variances in both a stock and especially a performanced engine. I cover these subjects in detail in Chapter II: Repairing the Achilles subsections, The S&S Solution and Moment Arm Torsional Bearing Wear and Knight Prowler Silent Belt Drive System.

Performancing the Twin Cam engine is at an advanced stage now since we have been able to work on them extensively for the last decade.

I am going to explore the many different and overlapping

ways to make your Twin Cam faster reliably. I will use products from Screamin' Eagle (www.harley-davidson.com), S&S Cycle, Inc. (www.sscycle.com), Rev Tech (www.customchrome.com), JIMS (www.jimsusa.com), Power Commander (www.powercommander.com), Terminal Velocity (www.terrycomp.com), Doherty Machine (www.dohertymachine.com), and a bevy of other venerable companies that produce superlative components for your engine. I will also show how to modify existing stock parts with a view to increasing performance such as modifying Twin Cam heads and ignitions.

If I describe some techniques that result in less engine longevity, I will explain this and why.

## Coast-to-Coast Riding Performance

However, my basic philosophy for street performance on

Evolutions and Twin Cams is that they will still be capable of reliable riding from coast to coast. This may sound like exaggeration or bragging but these engines are easy to make fast because many times I am only releasing horsepower and torque that is already in the engine but suppressed by emissions controls.

Furthermore, these are the first reliable Harley's that require less and less maintenance as the years go by.

I will give one example, chains. On a long trip, a rear chain needs adjustment at least once a day. It is common for chains to break on a long trip particularly the less expensive non-O-ring varieties.

Enter the early Kevlar rear belts with Aramid fibers and now the even stronger carbon fiber rear belts. Some riders will

go years without an inspection or maintenance adjustments checks and never encounter a problem.

## One Cubic Inch Should Deliver Minimum One Horsepower

There is a basic performance rule that an average H-D Evo or Twin Cam engine if built right should produce minimum of one horsepower per cubic inch at the rear wheel. Compare this principle with the anemic HP production of much less than one per cubic inch at the rear wheel in the CVO 110.

H-D does not put Twin Cam horsepower figures in their manuals anymore. They do still list torque figures, which in many ways are more informative than horsepower ones. However, the masses always use horsepower to delineate what is good, better or best.

The Twin Cam 88 has about 70 HP at the engine. By the

time, taking into account the frictional loss of horsepower as the engine power transfers back through the powertrain to the rear wheel the bike will lose around 14% of this. If the bike is out of tune, it will be more. If in perfect tune with some upgraded parts like a better grabbing clutch, it will be a little less.

Simple things like an out of adjustment clutch or improperly inflated tires or maladjusted primary chains and rear belts make all the difference in the world to what horsepower and torque is delivered to the rear wheel.

Let us now get back to the fact that superior engines built by the super wrenches easily exceed the figure of one horsepower per cubic inch. The 88 and 96-inch stock Twin Cams do not even come close to this figure and the 95 and 103-inchers even less so. Twin Cams generally produce less

than .8 horsepower per cubic inch at the

There is much horsepower to unleash in the new engines. I will describe different ways of letting the Twin Cam achieve better than the 1 horsepower to 1 cubic inch ratio at the rear Going beyond this may compromise the reliability required to ride coast-to-coast. Excellent mechanics can achieve up to about 1.2 or 1.3 horsepower per cubic inch at the rear wheel and still have a reliable bike.

A Current Horsepower Champ

However, there is an impressive guy by the name of Wolfgang Grasser who consistently wins the Daytona Brute Horsepower Shootout. He has done it again in 2008 with his 98 cubic inch Twin Cam using Wood Performance (www.woodcarbs.com), Knight Prowler TW-75G Cams and a whole bunch of other stuff.

Wolfgang's naturally aspirated 98 cubic inch Twin Cam produces an amazing 171 Horsepower! This is an astounding 1.74 horsepower per cubic inch at the rear wheel.

A very reliable street bike can easily up the anemic stock .8 horsepower per-cubic-inch at the engine to one horsepower per-cubic-inch at the rear This is what I aim for on my own bikes because I also want a lot of reliability.

## The Out to Lunch Bunch Salespeople

In my humble opinion, the salespeople that push first the 95-inch upgrade by itself from 1999-2006 and now the 103-inch upgrade for the TC96 are out to lunch. Most do not understand performance or the performance parts they are selling. They are clueless that one performance change leads

to another, as support components must assist the change.

Yet the trusting consumer eats it up. In my opinion, a strong manufacturer brand will cover and forgive many mistakes down the chain. We at Heavy Duty Cycles (www.heavydutycycles.com), and those that know what they are doing in other shops and dealerships must deal with setting bad information straight every day. There simply are not enough competent people that have a solid technical background to staff the many dealerships during the Harley-Davidson boom.

In fairness, to level the playing field, I must also say that the many aftermarket shops have an identical problem with performance advice and with custom bike building. Heavy Duty Cycles could spend half their service time out back just correcting the amateurish and many times dangerous mistakes

made by some of these cool chopper shops that seem to spring out of nowhere who promote themselves as the latest and the greatest.

A cool hairdo and brand new tattoos do not equate with knowledge, or does wearing Harley-Davidson apparel at a dealership. Whether building a street reliable fast hot-rod engine tailored to the customer's personality or designing and constructing a safe mechanically sound ground-up custom, the people doing it must first have an aptitude for the work and then many years of experience and technical expertise.

Ill informed sales people without adequate technical backgrounds tell trusting customers ad infinitum on what to do to their new Twin Cams.

The 95-Cubic Inch Upgrade

Everything I say about the TC88 to TC95 big bore upgrade equally applies to the TC96 to TC103 upgrade, as they are the same thing.

The most pervasive example of sales staff up selling that immediately comes to the forefront is the revered 1550cc upgrade kit. Yes, the manufacturing of the Twin Cam is superb. The big-bore kit increases engine displacement by 100cc's from 1450cc to 1550cc relatively inexpensively while the TC96 to TC103 increases 106cc from 1584cc to 1690cc. This upgrade is very reliable since increasing cylinder/piston bore (diameter), size for more cubic inches does not reduce engine longevity.

So, what is wrong with this picture? Where are the ill-informed sales people screwing up? The salespeople want to sell accessories and performance parts for their new bike sales

because many receive commissions and make a piece of each sale. Sometimes they sell parts that give good value and sometimes they do not. Sometimes they have rudimentary knowledge and sometimes they do not.

The Screamin' Eagle air cleaner is an example of getting good bang for the buck in the early days. This item adds a couple of ponies at the rear wheel so it is inexpensive horsepower. This allows the engine to breath better. Horsepower unleashes that emission controls suppress.

Salespeople sell this item to this day on almost every bike whether the customer initially wants add-ons or not. However, now that the Twin Cam is maturing and the aftermarket has caught up, there are just so many air cleaner systems that give more performance for similar amounts of money, the Doherty Power PACC being a great example, (www.dohertymachine.com).

# The Harley Lung

The internal combustion engine like Harley-Davidson may compare to the human lung. The more air that ingests and expels determines engine efficiency and power. The volume of intake air through filter, carburetor or EFI, manifold and intake ports determines how much gas can burn and therefore, how much expanding heat generates to drive down the piston on the power stroke. The volume of the spent exhaust gases must remove from the combustion chamber on the exhaust stroke, pushed out by an ascending piston and the principles of scavenging. The useless (except with turbocharged engines where the gases drive the turbo impellers), exhaust gases must exit the exhaust valve; exhaust port and the exhaust system. If they do not exit in an efficient or timely fashion, the virgin charge of gas and air entering the

combustion chamber will bastardize. The lung will suffer with emphysema and not breathe efficiently.

Performance is usually associated purely with speed but street performance can also give better gas mileage and produce fewer pollutants. In any internal combustion engine, not all of the fuel burns. Big proportional gains in speed are achievable by burning a higher percentage of the gas that is already there! The by-products of more burn will also be improved mileage and the production of less harmful emissions since more fuel burns.

I will be looking at Screamin' Eagle and aftermarket component cams, ignitions, pushrods, valve springs, clutches, carburetors and EFI. Moreover, I will examine the benefits of porting and polishing headwork in addition to eliminating the troublesome Twin Cam timing chain apparatus.

Gaining additional reliable power is straightforward and easy for those in the know; now with the extensively and continuously developing aftermarket and Screamin' Eagle product lines resulting in a superb line-up of parts and options for every taste. However, there is quicksand on the information curve that restrict the bike owners choice to good accurate information in order to make an informed decision on what is best for him or her.

## The Air Cleaner

An analytical mind will soon realize that the Screamin' Eagle air cleaner kit is a K&N air filter that allows more air through it with a modified back breather-support-plate. The thinking mechanic will cut out areas in the backing plate, outside the circumference and around the filter, again to let more air be accessible to the filter. Therefore, a competent mechanic can easily modify the stock breather assembly to

give as much extra power as the early Screamin' Eagle assembly. However, Screamin' Eagle gets the credit for thinking of this particular innovation first. I extensively cover air cleaner design and air filters in Chapter IX: Fuel Delivery, subsections, Air Filtration and Power, Velocity Stacks, Cleaning Air Filters, Performance Filters and Air Breathers and Ram Flow Induction Systems.

## The Exhaust

Usually the next part change involves the exhaust. Whether it is with Screamin' Eagle slip-ons or other aftermarket exhaust systems, this change will add another 3 to 5-horsepower to a stock bike. The bike will also hopefully sound and look better to the owner.

A collector system will add more power than most other

systems. However, with most choices with the exception of drag pipes or the currently popular too long or too fat pipes there will be a gain. Drag pipes result in a torque/horsepower loss especially at lower rpms except on big inch motors.

As engine, performance gains increase; the type of exhaust becomes more crucial for optimum gains. A good exhaust system allows the engine as a lung to exhale all its gases with consideration for noise quality and concern for environmentally harmful emissions.

In short, what we are trying to do in achieving greater performance is to move a bigger volume of air in and out of the lung also known as the swept volume of the cylinder and compression chamber. More air goes in, facilitating a more complete combustion that makes the bike faster and have better gas mileage because more burns of what is already there. Burnt gases must leave to allow more fresh air in. A better-tuned exhaust, once again, removes a restrictive

emissions control allowing the release of existing suppressed horsepower.

So, what is the problem with ill-informed salespeople? It comes with the next most popular performance change, which is raising the cubic inches from 88 to 95.

Big-Bore

The thousands sell with no thought or knowledge of whether there is a support system for them to perform adequately. Well, adequately may be the wrong word since the 95-kits will give more performance in an otherwise stock bike than the stock 88-incher. However, the question becomes; is it giving the total performance it would be capable of with a complementary support system? The short answer is no, it does not. The long answer is that the Twin

Cam engine does not have the lung capacity for large performance gains unless addressing breathing in a big way. More correctly, the lung is too big for the port throats that draws in the air and expels it.

The early stock Twin Cam has 1450cc (88 cubic inches) of lung capacity while the upgrade 95 cubic inch kit translates approximately into 1550 cc capacity. The Twin Cam lungs cannot operate at peak efficiency without increasing the breathing capabilities.

Inadequate Head Flow

Who was to know that the visually superior 1450cc Twin Cam heads would flow less air than the 1340cc Evolution without testing back in 1999? One would think that the first area that most salespeople would address in advising on how to make customer Twin Cams faster would be breathing since

the lungs (cylinders) are already too big for the air passages.

Why would H-D make heads for a larger engine like the TC88 that would flow less air than a smaller engine like the Evolution? This becomes even more ludicrous when most riders and dealers seek to increase the 1450cc lungs to a capacity of 1550cc as their first choice of going inside the engine as a performance upgrade. Those that recognize this incongruity and address them will make the fastest and most reliable engines.

The cylinder head design itself would fool anyone at first glance. Visually they seem superior. In design, they are but not in the quantity flow of air for the cylinders, they must try to fill. These heads, on a smaller cubic inch engine would kick butt.

Everything is there visually to inspire confidence in their efficiency including a bathtub shape combustion chamber and 60% more finning on the heads and barrels to increase the air-cooling. The TC88 bathtub combustion chamber shape we are familiar with on hi-performance aftermarket Evo heads leads the observer to draw wrong conclusions as to head flow.

The TC88 bathtub chamber is small having only 85cc volume, which is why the compression ratio is higher at between 8.8 and 9 to 1 compared with the Evo 8.5 to 1. The TC96 combustion chamber increases compression to 9.2:1.

With bathtubbing, there is a flat squish band all around the chamber so that the big round 1450cc, 3 ¾" bore flat-topped piston squishes the air/gas even more turbulently into the smaller chamber. This promotes superior flame travel and the highest percentage of burn on any Big Twin to date. The permanent mold TC88 head has the same size intake valve as the Evo but the exhaust is about .030" smaller in diameter.

The valve stem diameter and length of the Evo and TC88 exhaust valve is close to the same so it is very easy to put in the .030” larger V2 exhaust valves into the early Twin Cam heads to increase the flow capacity.

This should be a standard and easy partial fix whenever the heads are off for the first time.

Replacing the exhaust valve with a .060’ bigger one in the aftermarket will help further but not solve the main problem. There are valves up to .100” bigger for the intake. Again, this will help but not be the solution. The TC88 head flows less air than an Evo head, this because of the smaller intake and exhaust tracts. If the lung capacity were less than an Evo, say equal a 1200 Sportster, then these heads would offer the best performance possible because their design and capacity would be match to cylinder fill.

Surely, part of the reason for the stock TC88, flow deficient heads is a pollution control. The design facilitates burning more of the harmful pollutants.

However, one cannot ignore the monetary reasons of attempting to wrest back the aftermarket for their own gain. Hence, H-D, for the first time began manufacturing a full complement of aftermarket parts like superior flowing heads for their base platform product.

## A Lighter Valve Train

The Twin Cam decreases the valve stem diameter from ⅜” to a thinner and lighter diameter circa 2004.

Valve trains in pushrod-operated engines like all modern Big Twin Harley's are heavy affairs to move up and down with rpm speed. Valve train weight limits rpm capabilities. Overhead cams like those used in the V-Rod that actuate the valves directly are capable of high rpm's.

Any time the valve train can lighten, the engine will be capable of slightly higher rpm and therefore more power. There will also be the added advantage of increased parts longevity. A lighter valve train results in less necessary valve-spring-seat-pressure to return the valves to their seats as the commanding valve lobe rotates to a lower position. This relieves pressure on the valve train allowing parts to move faster and to last longer.

Porting and Polishing

Porting and polishing the stock heads works very well up

to certain levels of performance. This is necessary as a first internal engine step on performancing the Twin Cam with a stock displacement and especially before increasing to 95 or 103 cubic inches.

Dealers and consumers that increase displacement first have it all wrong. Yes, the 1450 will become a little faster when it transforms into a 1550 but not anywhere near its potential. The same holds true for the TC96 increasing to 103 cubic inches.

Increases in displacement either require porting and polishing the stock components or going to aftermarket heads. Cost is a concern as the porting and polishing with bigger exhaust valves and high lift cam valve springs costs less than replacement performance heads do.

One would logically expect the bigger TC88 engine to not only have bigger valves but also more intake port and exhaust port volume capacity. Not so, as the TC88 intake port volume is a few cubic centimeters smaller than the Evo.

Most of the porting work as usual will be near the valve seat. This is where most of the horsepower gains achieve on the Evo and the same will be true for the Twin Cam. Except for right under the valve seat area, the TC88 intake tract is very similar in shape to the Evo.

One of the principle areas I address when porting TC88 heads is a severe overhang the valve seat relative to the head port. Valve seats are separate parts that press into the alloy heads, as a very hard steel alloy is required to withstand returning slam from the valves. For optimal flow of air, the valve seats should radius and blend into the port it serves.

On the 2007 models valve seat matching to the head ports improve from the 1999-2006 Twin Cams. While there is better matching of the seat there is regrettably still some overhang of the seat. This needs to be radiused for better flow.

I cannot over-emphasize this as most flow improvement in any porting and polishing operation achieves within ½" of the seat area. However, the type of porting requirements will differ. The Twin Cam intake tract narrows near the valve seat. In fact, it narrows too much being less diameter than the valve seat bore creating a bottleneck, so to speak. Put simply, the port must be radiused out to blend with the valve seat.

The Evo is the opposite, where the seat must blend out to meet the port wall. On both engine's, this small section of porting along with a fully radiused valve seat cutting, (up to 7 cuts where the Factory uses three), blending into the

combustion chamber will attain up to 75% of the improved flow characteristics.

At Heavy Duty Cycles, we spend many hours concentrating around contouring the tract where the valve guide enters into the port(s) and spend even more time working within ½" of the valve seat to accomplish increased flow.

Radiusing the valve seat is achieved by up to 6 cuts away from the seventh cut, which is at 45 degrees where the valve meets the seat. Stock valve cuts from the factory are only three, and The extra cuts in radiusing are rounding the valve seat with successive smaller angle cuts like adding and for easier and thus superior airflow.

Porting both intake and exhaust tracts involve reshaping and sometimes enlarging the ports to allow more flow. It is not necessary to polish the intake tract but it should be

smooth to unimpeded laminar fuel charge flow and velocity.

However, the exhaust port must polish to a smooth finish. We do not care what happens to these gases. The only goal is to get rid of them as quickly as possible. Polishing hastens their exit.

Porting and polishing stock manifolds give less than a 1% gain so it is not worth the expense unless on the drag strip where little increments can mean the difference between winning and losing. However, some companies might do this to a mirror shine that sell performance manifolds and may do the same thing to intake tracts on their head exchange programs. Why, they look prettier and therefore sell better to the unsuspecting consumer. It is all about money with some companies.

I cover headwork and the various aftermarket heads available in Chapter VIII: Eliminating the Performance Obstructions, subsections, Key to Power: Head Flow, Porting and Polishing, Cutting the Valve Seat, Valve Seat Fitment, The Flow Bench, Edelbrock Performer RPM Heads, Screamin' Eagle Performance Heads, and S&S Super Stock Performance Heads.

## Manifolds

The Screamin' Eagle manifold for carbureted versions is better than stock but in my view unimpressive. I prefer the Mikuni Smoothbore manifolds. However, I feel the S.E. manifold for the fuel-injected models is well worth the investment over the stock one.

Combine these with a slight bump in compression, some better gear-drive cams combining with the appropriate valve

springs and the Twin Cam is going to get very much faster.

The Performance Parts Team

Of course, a modified breather, carburetor/EFI upgrading, exhaust work and a non-emissions single-fire spark map ignition will increase the lungs operating capacity close to one hundred percent with a fully functioning, integrated team of performance parts producing up to a 50% increase in horsepower and torque. Assimilation of performance parts complementing each other allows each addition to add exponentially to the team effort.

Conversely, too big lift/duration/overlap cams for the intended level of horsepower increase will overwhelm the other support parts producing less power, not more.

For those that are on a budget we at Heavy Duty Cycles port and polish stock heads, utilizing larger exhaust valves and performance springs whose seat pressure and free travel match the cams we have chosen for the particular riders bike, weight and riding style. We increase the compression to between 10:1 and 10:25:1 by planing the gasket surface of the heads. Increased compression is very necessary in the power scheme of things. I cover compression at length in Chapter VII: Torque and Horsepower, subsections, Compression, Compression Equals Torque, Compression Ratio vs. Cam Duration, and Cylinder Pressure and Intake Valve Closing.

Moreover, if installing a 1550cc kit, the new pistons with a higher dome will also achieve to 10:1 compression ratio. Planing the heads allow going up a little or not going so much; therefore making it a more versatile choice in tailoring an engine to a rider's needs and wants.

Cams

As a gross generalization, high lift cams increases low-end power the most, compression midrange power and cam duration and porting and polishing, high-end power. Increasing compression has a positive effect on low-end torque as well as lowering the powerband.

However, cam lift influences the whole rpm range, as does compression increase. Cam duration increases power somewhat over midrange rpm and especially higher rpms.

Headwork and longer duration cams work best from midrange to high rpm riding when the physical Principle of Momentum known in the mechanical world as scavenging begins to produce its magic.

Plane the heads .055” for increasing compression to 10:1 from the stock 8.8:1 if staying at 1450cc. Why go to the greater expense in buying new higher compression pistons when the stock ones will do. If going to a 1550cc big bore, increasing the compression accomplishes easily by planing more or less than to complement the other team players of performance parts.

Do not write this figure down as a magical one because the coming paragraphs will show that it must often vary depending on parts combinations used.

The correct way of determining the amount of planning required is to use the formula:

$$\frac{1}{1} - \frac{1}{1}$$

$$\text{Mill Amount} = \text{engine stroke} \times (\text{stock compression} - 1 \quad \text{desired compression} - 1)$$

This can also modify slightly by the thickness head gasket chosen by the builder. The crushed thickness of a Twin Cam 88 head gasket is .052”. Gaskets are available from the aftermarket such as James Gaskets, Cometic, and Accel to name a few with crushed thickness dimensions like .059, .045” and .032”. This allows the builders the flexibility to modestly increase or decrease the compression ratio of the Twin Cam. Gasketing will not replace planing heads or higher compression pistons as the gaskets modify compression up to a half point.

Matched Cylinders and Pistons

In 1999, I thought selling matched pistons to cylinders separately to be a dangerous practice and preferred to bore the cylinders to fit the new pistons to ensure correct fitment. Well, I must confess that after checking hundreds of these independently matched cylinders and pistons that I have only found one set out of tolerance. That does not mean for one second that I do not check them each-and-every time we install new cylinders and pistons.

## TC88 and TC96 Cylinders are the Same

It is important to realize that the stock TC88 bore cylinders are the same castings as the 95” barrels, which are the same castings as the stock TC96 cylinders and the 103” barrels. The 95” cylinders are bored out at the factory from TC88 cylinders as are the 103c.i cylinders.

Both the TC88 and the TC96 cylinder bore is 3 ¾” and when displacement increases to 95 cubic inches and 103 cubic inches respectively, their bore increases to 3 ⅞”.

Shops that have the correct boring bars, precision hones, measuring tools and torque plates can easily bore out the stock 88-cylinders to the new 1550cc pistons. This is a cost saving over buying factory-bored cylinders.

Torque plates are ½” to 1” thick so they won’t flex, perfectly flat steel plates that fit over either ends of the cylinders on the base and head gasket surfaces. They connect with long studs that torque down in the same sequence to the exact values as when installed on the bike. The cylinders are then bored under actual operating conditions to ensure the bores are straight and true. When removing the torque plates after boring, the cylinders will spring back to their free shape, which will be out of round and not true. When installing the cylinders on the engine with the new pistons, the cylinders

will revert to a true concentric shape when torqued down to specifications.

Exhaust

Mufflers or complete exhaust systems have three necessary qualities they must fulfill for the thinking rider.

1. Many consumers only go for good looks in order to personalize their ride and to individualize it from other bikes. This is legitimate because who amongst us does not want to look good.

2. The second reason is a pleasant sound to the rider's ear.

3. However, we should all consider a third factor above

aesthetic and sound quality.

Performance is free if choosing a performance exhaust in conjunction with the other two criteria.

Even if performance is not on the list, a more efficient breathing bike should be. The combustion chamber is like the human lung where any impediments to breathing restrict engine efficiency and its corollary performance.

Many exhausts on the market fulfill all three goals for the discerning rider.

Screamin' Eagle discontinued its previous line of off road mufflers. Their new line only fits the latest Harley models. They are street legal, wherever sold, on stock or mostly stock bikes. The S.E. mufflers use the stock header pipes and all emissions fittings. They also retain the crossover pipe.

Harley accomplishes street legal performance add-on pipes by changing the muffler diameter and internal baffling to filter out higher frequencies and accentuate the lower tones that are popular with riders. The pipes still result in about a 3% torque increase and up to an approximate 7% horsepower increase on a Twin Cam 96. Of course, complete performance exhausts like a collector system will up these figures considerably.

I currently use the stock FLT model headers on my Streetglide. Mufflers need big volume with a restrictor pipe at the rear the same diameter as the header pipe. I chose the S&S mufflers for my 2007 hopped up TC103 Alpha engine even though the Screamin' Eagle mufflers produce good power while staying within EPA noise regulations and meeting emissions standards for new bikes.

Let us take a quick look at TC96 Dyna; FLT and Softail torque figures utilizing stock exhaust systems. The Dyna produces 92 ft-lbs of torque at 3000 rpm, the Softail 90 ft-lbs of torque at 2750 rpm and the FLT produces the most torque of the three with 93 ft-lbs at 3500 rpm. The Softail comes in last for two reasons; one there is a parasitic loss of power because the engine must power the counterbalancers and secondly the muffler sizing is inadequate. The Dyna does not have to contend with counterbalancers but it also has too small volume mufflers. The FLT has larger volume mufflers and thus produces the most power of the three.

These S&S mufflers are bigger volume than most other aftermarket ones and therefore more capable of handling and accentuating the extra ponies and torque. They also have a hidden restrictor pipe, necessary to flow out the spent exhaust gases efficiently. These oval pipes look weird by themselves; they are so big, when viewed on an FLT model without the

saddlebags. Once installing the saddlebags, these stainless steel over-size mufflers hide under them, and they have cool appearance protruding slightly behind the bags.

The reader only needs to look at the really fast performance Japanese, European or our own Buell motorcycles to learn how to mimic what the really fast bikes use for mufflers, which is a big volume expansion chamber with a cap on the end to house the much smaller diameter outlet restrictor pipe.

Exhaust Reversion

I cover reversion in many other sectors of the book. Proper exhaust reversion is all about ensuring that returning Kinetic pulses of energy returning back up the exhaust does not reenter the combustion chamber. This accomplishes by

exhaust pipe diameter, length, muffling, torque cones or anti-reversion steps in the heads.

## Recalibration

In my opinion, the rider should view recalibration with a jaundiced eye. My view is that recommended recalibrations, burns or downloads are not always necessary on both previous and present Twin Cams and Sportsters for that matter.

In fact, on tuning thousands of Harley's at Heavy Duty Cycles on both carbureted and fuel injected engines, my experience is that the recalibration of EFI or the rejetting of a carburetor is totally unnecessary with many changes especially with exhaust and air breathers. This is because many Twin Cams deliver from the Motor Company with a rich air-fuel-ratio (AFR). Minor performance changes grow into the

rich condition and lean it out to a correct AFR,

However, never ignore the possibility an adjustment may be necessary. We check every bike we work on to see if requiring an adjustment or not

Furthermore, in my opinion some stage I kit recalibrations for EFI are nothing more than an rpm limiter adjustment. I recommend using the 1550cc stage I recalibration kit on the 1450cc stage one kits, which provides a tuning function as well as an increase with the rev limit. The tuning function increases gas delivery thereby eliminating popping and flat spots.

Do not use this trick on the 2006 Dyna or 2007 and later Twin Cams, as it will not work properly with bikes with 02 (oxygen) sensors. Listen to a knowledgeable tuner that only

up sells when necessary. Patronize the tuner/mechanic/ technician that will assist the rider through this sometimes-confusing world. Pay attention to the mechanic you learn to trust.

TC96 Head Port Improvements

Another TC96 improvement to the reworked head castings is that a restrictive-to-flow curvature on the inside of the turning port eliminates with the over sizing of the whole port but still radiusing narrower down to the almost same size valve seat I discussed above. This narrowing down will increase flow as explained by Bernoulli's Principle.

Sir Isaac Newton explains the movement of a fluid or gas through a pressure difference. Suppose a fluid is moving in a horizontal direction and encounters a pressure difference. This pressure difference will result in a net force, which by

Newton's second law will cause an acceleration of the fluid.

I measured some 1999-2006 earlier exhaust head ports and found an average approximate diameter is 1.330" (this varies from head to head). The new head is about 1.510". This increase in sizing is huge as thousandths of an inch are equivalent to miles inside an engine. Intakes are pretty much the same size but a noticeable; and much improved difference is that the seat edges and casting they fit into machine to mate more correctly on the heads I removed from my 2007 FLHXI.

The head casting is nicer with smoother edges and more defined such as oil cavities etc. although it really does not affect anything mechanically; it is just nicer.

The valve seal improves from the 2002 upgrade

(18094-02), which was also a good one. It incorporates into the lower valve collar. The seal looks like to be a better more pliant rubber-like material that will have a longer service life. The new valve seal is orange in color (18094-02A) while the 2002 to 2006 (18094-02) are black rubber.

The heads still need porting and polishing to increase the horsepower, which did not increase much from the Twin Cam 88 to the Twin Cam 96 even with eight more cubic inches and higher compression. However, the TC96 head is an improvement over what went before.

The TC96 Longer Stroke

Compensating for the stroke increase from the TC88 4.000” to the TC96 4 ⅜” are lighter pistons, wrist pins, and the small ends of the connecting rods. Less weight will offset

the effects of the piston traveling a greater distance up and a corresponding greater distance down the cylinders in the same time-period. There is less mass that must change direction at TDC and BDC.

The TC96 engine can also increase to 110 cubes with a 4.000” cylinder bore and 113 cubic inches with a bore of 4.060”. It is more complicated and expensive to achieve these bigger engines since they require machining out the cylinder spigot holes in the engine cases to accept these bigger bore cylinders. However, it is not necessary anymore to purchase bigger (stroked), flywheel assemblies since we already have them in the stock TC96.

Pistons

The TC96 compression ratio increases to 9.2:1 from 8.8:1 on the TC88. This will add some performance to the engine,

particularly low-end torque. Initial increases to compression have much more impact with performance than subsequent raises. In other words, the first bump gives more bang-for-the-buck than later or additional ones.

Each piston's greater swept volume is responsible for four more cubic inches into the same size combustion chamber used since 1999.

The swept volume or engine displacement is due to each piston traveling further up the cylinder and lower in the cylinder for a total of ⅜" more travel, which of course equals the stroke increase from 4.000" to 4 ⅜".

I will discuss the pistons more as they relate to less reciprocating weight next.

To help reduce the longer-stroke, larger-displacement engine's vibration, the engineers lowered the flywheel assembly's reciprocating weight by designing new connecting rods and pistons. They are lighter and supposedly without strength reduction.

If the reader has a look at the connecting rod small ends that connect to the piston wrist pins, he or she will probably take issue with the above statement. There is a picture of a TC96 connecting rod small end in Volume I: The Twin Cam in Chapter VIII: The 2007 TC96 In the same volume in Chapter I: March 1998: The Introduction of the Twin Cam there is a good picture of a TC88 crank that shows the beefier con rod small ends. The connecting rod small ends taper on the 2007 to present Twin Cam models, taking away much material away when comparing to the squared full size 1999-2006 connecting rod small ends.

There are two groupings of parts whose weight is important in engine crank balancing in order to reduce vibration. They are reciprocating and rotating parts. Reciprocating weight refers to the weight of all the flywheel parts that go up and down the cylinders like the connecting rod small-ends including the pistons, wrist pins, clips, and rings. The other component in engine crank balancing is the rotating weight. These parts do not go back and forth but spin or rotate like the flywheels, shafts, crank pin and bottom end connecting rod big ends and bearings.

The TC96 small-ends may be strong enough but in my opinion cannot be as strong as the TC88 small-ends they replace. The connecting rods have tapered top ends, while the pistons redesign to be like many stroker pistons.

The wrist pin bores are closer to the oil ring lands, which results in a shorter piston and 10% less weight for a 6-percent total reduction in the flywheel assembly's reciprocating weight. The piston wrist pins are also shorter than TC88 wrist pins and therefore lighter.

As with any piston, the wrist pin is in the middle of the piston parallel with the piston front and rear thrust faces. The thrust faces are round to match the round cylinder that the pistons move against separating by a film of oil. The areas between the TC96 thrust faces under the ring lands are flat and move inwards since there is no contact with the cylinder bore resulting in a lighter piston. No H-D Big Twin and XL pistons that have gone before contact the cylinder bore on the non-thrust faces. However, they do not move inboard on the non-thrust surfaces. These earlier pistons have an elliptical shape that the reader probably would not notice unless looking for it. The front and rear thrust faces are still round but the sides move in slightly.

The pistons change direction slightly as they go up and down the 45-degree opposing cylinders so they need to be able to rotate slightly on the con rod bushing. The wrist pins usually fit into the piston bosses with a light hand press at room temperature.

The less mass or reciprocating weight that must move up and down the cylinder will reduce internal horsepower losses but also relieve stress. Imagine a piston barreling up the cylinder when suddenly in an instant it must stop and change direction. The same thing happens at the bottom of the cylinder. This racing, stopping, changing direction, racing, stopping, and changing direction occurs thousands of times a minute (rpm). With a stroke increase, the speed at which this happens also amplifies since the piston must travel further but do so in the same time-period.

The advantage of the stroker is more power. The disadvantage is the piston speed increase with a shorter piston and resultant generating heat. The disadvantage is offset but does not cancel by the lighter piston, wrist pin and connecting rod small end and consequential less reciprocating weight.

The increase in TC96 stroke also gives a favorable con rod ratio thereby increasing low-end torque.

Timing Chest Improvements

The successful to date timing chest changes experimented with in the 2006 Dynas have all been incorporated into the new Twin Cam 96 Big Twins.

The three boondoggles to the Twin Cam engine efficiency and performancing have always been insufficient head airflow, problematic timing chest mechanical deficiencies, and emissions controls. The heads can easily improve with to allow more air through to service the gasoline-mixture combustion needs of the engine.

However, H-D is really getting up to speed with the emissions and performance benefits of electronic fuel injection (EFI) and the sophistication of engine-management-system computerization that I will also explore in this chapter.

Some timing chest changes seem to be very much better and will eliminate some of the problems in the past.

The new roller chains do not have sharp edges like the 1999-2006 silent cam chains. Cam drive chain failure on these engines has not been an issue as of yet. These new roller

chains combine with new tensioner pads that operate hydraulically to replace the troublesome and unpredictable 1999 to 2006 spring-loaded tensioners. However, I am beginnining to hear unsubstantiated rumors since I have not seen this myself of later hydraulic chain pad wear.

## The Cam Support Plate

It seems that every time I see functional corrections to a problem like the hydraulic tensioners and rounded edge roller chains, I also see the opposite. This will be a component that was working well screwed up. The cam support plate replaces with one that utilizes parent material cam-support-bearings instead of the previous roller and ball bearing. I hope I am wrong about the longevity of these parent material-bearing supports that I discuss at length in Chapter II: Repairing the Achilles Heel, section, Moment Arm Torsional Bearing Wear.

I recommend replacing the timing chains and cams on all year Twin Cams with a gear-drive-cam-system especially with any performance work.

- The gear driven cams do not have the same torsional loads on them after eliminating the timing chain apparatus.

- The gear drives also eliminate any cam-timing variances that may occur with the timing chains.

- Finally, the gear drive cams will dominate any performance valve springs to ensure they perform their jobs competently.

In other words, stronger seat-pressure springs cannot overwhelm the gear drive as they sometimes do with the timing chain adjusters.

The Oil Pump

The new 2006 Dyna and 2007 to present Twin Cam oil pump is a tremendous improvement over the previous 1999 to 2006 one. The earlier oil pump is efficient and works competently. However, the new pump improves scavenging capacity by 23% and feed by approximately 10 percent.

This scavenging boost is necessary because of the increase in engine displacement of eight more cubic inches from the TC88 to the TC96. The 1999 to 2006 (excepting 2006 Dyna), pumps have a 12/13-lobe interaction. The outer geroter has 13 internal lobes while the inner geroter has 12 external lobes. Therefore, with the new 2006 Dyna and all 2007 to present Twin Cams 10/11-nub rotor arrangement, more oil feeds into the engine. Accordingly, more oil removes from the

timing chest and returns to the oil tank instead of carrying up into the top end, driven by the ongoing voluminous air displacement in the engine.

This air movement occurs every time the pistons descend in their cylinder bores. The oilified air forces into the timing chest cavity and up through the pushrod tunnels sometimes over-whelming the breather apparatus in each head. In a perfect world, the breather apparatus separates oil from air so that the air alone can vent to the outside world. Certainly, the new oil pump removes more oil from the engine, which should prevent some oil venting up into the breather apparatus. Sometimes, too much oil still overwhelms the breather apparatus but the new pump is a big improvement, nonetheless.

Additional help for breathing such as the S&S Reed Valve (www.sscycle.com), the Doherty (www.dohertymachine.com), Mystfree breather or the TP Engineering

(www.tpengineering.com), ProVent rocker boxes is often necessary especially with an increase in cubic inch displacement.

The increase in swept volume means that more air displaces in a bigger engine bringing more oil into the rocker boxes for separation. Blowby can be problematic in these big engines, so both the new oil pump and an assist by the aftermarket is usually necessary. For further reference, I discuss this situation at length in Volume I: The Twin Cam, Chapter IV: Harley Oil Leaks: Marking its Spot, section, Defining Blowby, Oil Carryover, and Venting.

## Building an Engine the Wrong Way: The Saga of R'dR

Once addressing the three Twin Cam bottlenecks to power, which are:

1. Insufficient head flow,

2. The timing chest,

3. Emissions Controls.

One fast Harley will be the result.

A not-so-recent customer by the name of R'dR brought a very expensive hi-performanced Harley motor to us at Heavy Duty Cycles. R'dR liked to ride a lot but the engine didn't run right; was very temperamental and broke down a lot despite all the money spent. He is not a rich guy but had put at least three times the money into this engine than what it cost new. His frustration was affecting every facet of his life. R'dR's wife told him that this was the last time he was going to get the engine worked on. If it did not work properly, a divorce was

in the offing.

His previous methodology in having his dream engine built follows what many riders do. R'dR read magazine tech articles about the individual superstar parts available for his engine.

I will not get into here that some magazines will print whatever an advertiser wants in order to land the advertising account. There are two major reasons I write for American Iron.

1. One, it can be left on a coffee table with kids around.

2. The second is that the editors make every effort to print accurate technical information. The editors have been around bikes their whole lives, it is their passion, and they know their tech.

Anyway, R'dR, in good faith, put the best heads, the best cams, the best carburetor, the best exhaust, the best stroker flywheels, the best big-bore cylinders, and the best ignition on the bike.

## R'dR's Downfall

So what was R'dR's downfall? He and his many advisers did not coordinate the parts so that they supported and acted in conjunction with each other. A good analogy is sports teams win championships, not by having a superstar hogging the ball but by every player being a component part of the success. All of the parts were arguably the best in each article describing them.

Many believe that nothing makes power like more cubic inches thereby increasing engine size by creating more swept

volume. More area allows for more air and gas, which allows for increased burn that in turn causes more pressure forcing the flywheels to rotate by the descending piston on the power stroke.

R'dR wound up with one of the biggest cubic inch engines that stock Harley engine casings could support. I will dispel the myth that more cubes are the only way to gain more power. I will show time and again that if you burn more or a higher percentage of the air and gas that is already in the cylinder that the same net effect is achievable.

In physics, burning fuel creates heat, which is the same as pressure since heat expands. The amount of pressure exerting on the downward piston is all about performance. The old timers whose mantra is more cubic inches do not totally understand the new technology, combustion chamber head

design. They achieve power by putting a lot more air and gas into bigger cylinders created by an increased bore and stroke.

For the most part, they do not concentrate on increasing the percentage of the fuel burning. Mechanics that have a good understanding of percentage of burn and flame travel achieve more power by utilizing more of what is already there. Primarily, newer mechanics learn this stuff, because of emissions control laws. The additional benefit to their work is the engine emissions become environmentally friendlier.

The really good performance mechanics combine all the safe ways to increase power instead of just relying on one because of limited vision or knowledge. Big bore with safe strokes that combine with superior head design and sophisticated ignitions are the territory of the thinking mechanics.

Parts replacers are under-achievers in this fascinating world of power.

R’dR’s Performance Components

Replacing the solid-state computerized engine management system was a mechanical points/condenser ignition.

“What?” I exclaimed.

“A racing ignition,” says R’dR. A beautiful and very big carburetor had replaced his fuel injection.

“Why would you put a carburetor on a bike with Delphi EFI?” I asked incredulously.

“And points, racing or otherwise? I do not get it R’dR.”

“Well you can work on points and carburetors and they are adjustable,” says he.

“You don’t need to adjust anything with electronic ignitions or EFI once they are set up,” said I.

Carburetors are rapidly becoming outdated because of their inefficient way of delivering fuel. This inefficiency allows more pollutants to escape into the air we breathe. If a carburetor mixed and delivered fuel more efficiently, a higher percentage of the fuel would burn creating more power with the added feature of creating less pollution. This is the simple proof the carburetor lovers refuse to understand never mind acknowledge.

Increasingly effective ignitions help out the poor old carburetor in burning more of the fuel but if not there at the right time or in the proper ratio of air to gas, the hottest spark in the world will only be of minimal help. Let us compare spark created by the now antique but much beloved by many, points ignition system that deliver only one weak spark at a time. I say weak because the points/condenser system only can deal with at most, only half the voltage of even the worst modern electronic ignition. Weak also comes into play because so many things can go wrong with the spark with condensation or oil soaked points surfaces that do not transmit electricity properly not to mention out of adjustment or corroded points.

Crane Carburetor Electronic Single Fire Ignition

Compare this points/condenser system with a Crane

(www.cranecams.com), HI-4TC bolt in electronic ignition that I use in conjunction with carbureted Twin Cams.

The HI-4TC Twin Cam ignition replaces the popular and functional Crane HI-4 ignitions I use on all 1970 to 1999 Big Twins and Sportsters. The first thing the Crane ignition does is eliminating existing emissions controls that restrict power. The stock electronic ignition's main purpose is to reduce pollutants with no regard to how this affects power. This is the first component of a good ignition. I am not being disrespectful of the environment. As I have said many times, power is created by burning more of what is already there, which also means that pollution controls in a different way.

Easily eclipsing the maximum 20,000 volts that generates by a points ignition system in perfect tune is the minimum 40,000 volts generated by any electronic system. The single-

fire premium electronic ignitions like Crane's develop even more voltage than this. Think for a moment how much more gas the extra voltage crossing the sparking plug's electrodes will burn. Furthermore, the gap between the spark plugs electrodes can double because the extra voltage can jump more distance. We now have more fire and it is extending double the distance through the flammable fuel. This should be enough to convince even the diehards that replacing old technology is necessary.

However, it is much more than this.

The single weak points spark or the single stronger electronic spark generating by stock electronic ignition systems replaces by many Crane ignition features. Up to 9-sparks at idle and 3-sparks up to 6000 rpm through a process called fast inductive re-strike on the Crane HI-4TC ignition system. The spark programs to start on the first turn over of the engine.

A digital rev limiter allows 100-rpm increment settings from 1500 rpm right up past redline. The stock ignition only programs in one position to protect the engine from mis-shifts that potentially rob peak power. The Crane adjustable system allows a rev-limit to be set just before the powerband peaks.

The powerband shifts whenever performance work occurs. The trick is how to shift it to where you want for maximum performance coinciding with a particular riding style. The Crane ignition rev-limit feature allows for any type of powerband. However, I would never set it below 5000 rpms and never, never anywhere near the 9900 rpms capability of the Crane ignition.

A pushrod-operated engine simply cannot handle that fast an rpm, since pistons would keep going instead of reversing

motion at the top of the stroke. Pushrods would bend and self-destruct while the valves would attempt changing holes.

The overhead cam engines like the V-Rod can easily handle this high rpm. This is an indicator of their superior performance technology. In addition, overhead cams demonstrate how fast V-Rod's are out of the crate.

Adjustable initial timing is achievable with the Crane ignition. A potential problem with Twin Cams is that the timing is non-adjustable. This is one reason for stock engine detonation. Retarding ignition slightly to eliminate detonation is not possible.

The other reason is inadequate gas relative to air, which makes the engine run hot (lean). The Crane HI-4TC allows for a retard timing of up to –5 degrees crankshaft rotation where the spark goes off sooner or before the piston reaches top

dead centre (TDC). Allowable advance timing is up to +5 degrees. This is a big deal to a good running engine and even more to a performanced one. These timing changes allow timing adjustments from a stock engine to high compression modified ones.

Remember that compression creates low-end torque. Horsepower is torque multiplying by rpm. Increasing compression without an ability to modify ignition timing is a very dangerous game. Compression increases heat and thus power dramatically but without the proper ignition timing, detonation ensues, reducing engine longevity.

Each cylinder can time individually with the HI-4TC, which is a nice addition because the front cylinder runs cooler than the rear. For the sophisticated, there is a data link where a computer can program custom ignition settings.

LED Blink Diagnostics

Finally yet importantly, the Crane HI-4TC allows for the recording of engine data for further analysis. This diagnostic aspect is cool. The LED on the Twin Cam's instrument cluster comes on when the ignition switch turns on. It turns off when the engine actually starts if there are no internal problems.

If the Crane system detects trouble within the ignition's engine management system, the LED will blink anywhere from one to 7-blinks followed by a short 2-second pause.

1. One blink indicates a faulty crankshaft position sensor or a faulty connection in the sensor wiring harness.

2. Two blinks indicate lost cylinder spark synchronization causing a momentary loss of power or an intermittent signal

from the crank sensor.

3. Three blinks tell the rider that the manifold (MAP), pressure sensor has a faulty wire connection; a vacuum leak, a loose sensor or the sensor is defective.

4. Four blinks indicate that there is a low voltage battery problem. This can range from too many accessories on at the same time to a charging system defect such as a dead battery or burned out alternator and/or regulator.

How handy is this?

5. Five blinks is a high voltage battery situation, which point to an overcharging regulator, an almost dead battery, or faulty wiring connections. With electrical situations, always check the stuff like wiring connections that do not cost money

first. Many shops and riders diagnose charging problems by replacing parts.

This usually costs more money because of unnecessary parts substitution. The save money rule is to test the battery and charging system before reaching for your wallet.

6. Six blinks will designate a 2001 and later security system or bank sensor malfunction. The bank sensor turns the bike off if the bike tilts over more than it should as in tipping over.

7. Seven blinks deals with the HI-4TC's capacity to store engine information for analysis. It indicates that either the corrupt or lost information, improper programming by the data link or the unit itself is headed south. It is nice to have advance warning.

Not only is the HI-4TC a performance part that unleashes existing power and creates new power, it will also diagnose the engine management system; too cool!

So why did point's ignitions work at all, one might ask. Well, there were no pollution controls, gas was cheap, and it was all the riders of the day knew because it was all that was available.

However to stay a fan of points or carburetors is a walking statement of a lack of knowledge. However, for the tinkerers amongst us, we still love them.

To be fair, there are situations where carburetors may be the only choice for a particular application but I am talking about a reliable and faster street bike here. With this said and done, I also have fond memories and a special place in my

heart for points ignitions and carburetors. However, please do not try to tell me that point's, carburetors, or both will make your bike faster than a fully adjustable electronic ignition as part of a Delphi EFI engine management system.

I discuss performancing the EFI systems in Chapter IX: Fuel section, Performance EFI Ignition and Fuel The technology is a lot of fun and very impressive; much more so than the carbureted management systems that utilize units like the Crane HI-4TC. I also discuss the leading edge carburetor IST, self-tuning ignition from S&S Cycle, Inc. again in Chapter IX: Fuel Delivery, section, Intelligent Spark Technology.

Too Much Compression

I will now continue with R'dR's great intentions to develop a super engine that should have been reliable and real fast. He

had his engine cases bored out to accept a mega cylinder bore size that were designed by a company that supplies drag racers. The cylinders required special order pistons for blasting down the track.

Big bore is a desirable way to get more cubic inches since no potentially destructive increases in piston speed occur. This is why upgrades from 883 Sportsters to 1200cc, 1450cc Twin Cams to 1550cc and TC96 to TC103 cubic inches are perfectly safe and do not affect engine longevity whatsoever. The problem for R'dR is he did not have the information required to order proper pistons for street riding. The result was too much compression causing deadly detonation and rough running. In addition, because of their normal use mostly used on drag bikes, it became very difficult to find the right pistons commercially. Furthermore, the pistons cost an arm and a leg, as they were a specialty item, made from very good materials to attempt to withstand the abuses of racing and a diameter not common with the major manufacturers.

I kept the bore size on R'dR's rebuild because his engine casing spigots were bored to accept the mega-cylinders. If I put smaller, but still big, cylinders into the rebuild, there would be spacing between the outside cylinder wall and the engine casing spigot hole.

Some might say this would not be the end of the world but it invites side pressure on the cylinder hold down studs, as there would be room for the cylinders to shift. This is not the design function of these studs, which is to clamp down the heads to the cylinders and to the casings. Furthermore, the gap invites oil thrown up by the rotating flywheels to go into this area, which could be a factor in cylinder-base oil leaks.

In short, the rigidity and support by a snug cylinder fit into the casings would be lost.

## Too Much Stroke

R’dR was also ill advised on the best stroke for the flywheels, which determine how far up and down the cylinders the piston travels. Increasing stroke increases piston speed because the pistons have to travel further up and down in the same time-period as before. There are mechanical limits to increasing the speed as the pistons can only travel so fast before they start to wear prematurely and self-destruct.

R’dR’s supposedly knowledgeable articles and super wrench that advised him again resulted in flywheels being used with too much stroke for a reliable street bike. His special very expensive pistons began to wear too fast and the engine began to smoke oil out the exhausts. This is a sure sign of wear and the need for repairs.

Stroker flywheels have the mounting holes for the crank pin moved further out closer to the rims so that as the wheels rotate, the fixed length connecting rods that attach via bearings to the crank pin and to their respective pistons at the other end, move the pistons further up the bore and pull them down further an equal length.

Because a piston must move, further up and down than on a stock stroke bike, compensation is necessary so that it does not hit the head at the top or smash into the flywheel at the bottom. Using a piston that is shorter on both ends in order to avoid collisions at the top and bottom of the stroke normally is the route taken. Now we have a piston that is moving faster in order to cover more ground but there is not as much piston skirt area for support against the cylinder walls.

The stroker flywheels made by JIMS (www.jimsusa.com),

had superb quality but not designed for a reliable street machine.

The connecting rods were boxed I-beam constructed works of beauty. These connecting rods would take forever to wear out in a street bike and we quite happily put them back into the rebuild.

I guess the point that R'dR' missed is that the advertising and tech article claims for quality were very true except the design was for a very radical engine when he wanted some reliability with the extra speed capability.

The cams were excessively wild at over a .650 lift with mega duration and overlap to combine with the only weak link in a Twin Cam, which is the timing chain apparatus that drive the cams.

Cam Gear Drives

Stock cams are in the .475” lift range for both carbureted and EFI TC88 and TC96 Twin Cams. The radical cams require special valve springs that are much stronger. This plays havoc with cam timing since the timing chain adjuster springs are weaker.

At Heavy Duty Cycles, in go-fast engine modifications that deliver huge horsepower gains with no loss in reliability I routinely install cam lifts of .570” and .585” in TC88 and TC95 engines and use cam lifts of .585” and up to .625” in TC96 and TC103 engines. Of course, these cams must coordinate with other support parts such as cubic inches like the big-bore kits, higher compression, exhaust, headwork, and

ignition modifications.

For people that want modest gains, a .505 or .510 lift cams work great with free flowing breather and exhaust modifications. The difference in my cam selections is that I install the appropriate support parts that turn these cams from being a potential problem into very reliable setups that give oodles more power. I coordinate the appropriate valve springs that are stronger but not more than necessary to convey and carry out the wishes of the rotating elliptically shaped cam lobes to the valves.

I also convert the early problematic timing chains and the so-far reliable late timing chains to a gear drive similar to that used in pre-Twin Cam engines. The major difference of course is that the Twin Cam has two cams with two lobes each to activate two valves and most previous Big Twins have only

one cam with four lobes in-line that activates all four valves.

## Roller Rocker Arms

These cam lifts of .570” and greater that I use for specific engine combinations require the use of roller rocker arms. Again, a pushrod-operated engine has limits that are very risky to exceed.

In my opinion, the Twin Cam engine design simply does not permit such radical lifts as .600” for a street machine with timing chains for any length of time. As cam lift increases, the need for upgraded support parts like roller rocker arms becomes ever more crucial. R’dR either never read an article on roller rockers or his wrench neglected to put them in, which is unforgivable in the world of hi-performance. Roller rocker arms are a good idea if affordable even in a stock pushrod operated engine. However, as cam lifts approach .570

in the Twin Cam, they become necessary. Do not even go near the .585 lift or greater cams unless the installation of roller rocker arms are affordable.

Evolution engines need roller rockers at lower lifts like .525” because the pushrods are at exaggerated angles compared to the Twin Cam. There are many excellent manufacturers like Andrews, Crane, JIMS, and S&S.

As each cam lobe rotates to its highest position, the hydraulic roller follows its path over the lobe pushing the hydraulic lifter up. This in turn forces the pushrod up causing the rocker arm to rotate pushing down on the top of the valve stem to open the valve. More precisely, it rubs across the top of the valve stem in order to push the valve open. This side rubbing pushes the valve against the guide, which causes wear and resistance because of friction. Anything that can

lessen this side thrust is a big bonus in terms of efficiency and horsepower. Internal engine friction known as frictional horsepower loss lessens the available horsepower.

Of course, if the valve is at 90-degrees straight up and down there will not be any side thrust.

The roller rocker arms are superior to stock rubbing The tip of a roller rocker arm, which actuates the valve opening, is a roller supported by a bearing that pushes the valve open with minimal resistance. It rolls across the top of the angled valve as opposed to rubbing across it. As lift increases, so does side loading, friction and inertia, which leads to premature wear and a heavier valve train.

Tricking the Valve Train into being Lighter

Roller rockers along with low friction valve guides act in

concert to trick the valve train into thinking and reacting like it is lighter. This is very important as it relates to valve float with hi-lift cams and/or high rpm riding.

Valve Springs

R'dR had a bigger problem, which was the installation of wrong valve springs in his They were top quality and made by a very reputable manufacturer with which the reader would be familiar. The problem was that their design was for lower lift cams. Two of the springs had coil-bind and broken. The higher the cam lift progressively requires more seat pressure. In other words, the more radical the cam is, the stronger the springs must be, which translates into what I refer to as seat pressure. Coil binding occurs when there is not enough space for the springs to compress for the particular lift of the cam. R'dR's valve springs could not compress enough for the lift of

his mega cams. Their only option was to break in order to allow the stronger cam to rotate.

Usually a pushrod will bend instead of the spring breakage but there were super-duper chromemoly pushrods in the engine that will not bend for much. I don't use the ultra-strong pushrods in my shop because if there is a problem bad enough that something is going to break, I would rather a less expensive and easier accessible part become the victim.

When a valve spring breaks all kinds of stuff can happen like bent valves, broken valves and damaged pistons. As important as they are, valve springs are not the boss but a servant following the dictates of the rotating cam lobes. Stock valve springs may not be harmonically wound, progressively sprung or able to compress enough for the increased cam lift. Harmonically wound means that the inner valve spring coils opposite to the outer spring. Vibrations from the rapid movements will offset. They will hopefully self-cancel each

other.

Progressively sprung refers to the valve spring coils are wound progressively tighter going from one end to the other. This is very important as it helps prevent harmonic surging whereby the spring coils may fibrillate due to rapid changes in direction or movement.

Bolt-in cams refer to the fact that their increased lift is not enough to cause the valve springs to coil bind. Cam manufacturers like Crane recommend the strength of valve springs for installation with their cams. The figures are in pounds of seat pressure. Following these recommendations will allow the mechanic to install the appropriate springs for the lift cam chosen. Furthermore, this will spare the valve seats from too much slam since many mechanics overdo seat pressures and put too strong a spring in. If the valve returns to

the seat more forcefully then necessary then parts life will decrease.

Valve bounce may also occur by overdoing seat spring pressure. The valve is required to return to its seat too forcefully and bounces off it. There may be a rising piston in the way and parts will bend or break. The proper springs ensure against coil bind and valve bounce but will still prevent valve float under normal circumstances.

Valve float occurs when valve springs are not strong enough for the lift and duration of the cam lobes. It will also occur with a blown shift, which is why this can be injurious to your engine's health. In addition, of course there are riders that like taking their pushrod-operated engines into rpm areas beyond their design. After the cam lobes highest point opens the valve fully, the valve spring(s) must pull the valve back

into its mating seat as the rotating cam lobe diminishes in height. Valve float occurs when the valve does not respond right away due to a weak spring or extreme rpms overwhelm the spring.

A rising piston can crash into a floating valve that is in the wrong place at the wrong time. Valves bend very easily. When bent, they cannot sit in their seat properly and seal in the compression. No compression equals zero power.

Thankfully, R'dR's engine problems prevented much riding thus sparing impending doom from the broken springs

A good performance mechanic realizes that a lighter valve train has great benefits from a power point of view as well as allowing the engine to run more efficiently without as much power robbing and wear producing internal work. For example; lighter pushrods as long as they are strong are

important especially in hi-performance engines because they makes the whole valve train lighter, which means that there is less weight to move around. Less energy expends, which takes away from horsepower delivered the rear wheel. A huge benefit is that weaker seat pressure valve springs can also have use with lighter pushrods since there is not as much weight resisting the valves returning to their respective valve seats by the spring tension.

The Factory engineer's purposeful design in reducing valve angles in the Twin Cam relative to what went before is of immense assistance.

Lightening the valve train will allow conservative rpm increases above the stock 5500 rpm. I have a hard time saying this because I do not want to encourage riders to go into these higher ranges. As I said earlier, roller rockers along with low

friction valve guides act in concert to trick the valve train into thinking and reacting as if it is lighter. This is very important as it relates to valve float with hi-lift cams and/or high rpm riding.

Weaker springs, all of a sudden, perform the function of stronger seat pressure springs without the adverse effects of premature wear. The thinking mechanic always produces the best running, longer lasting and faster engine.

Late 2004 and all-later models use much lighter valves as the valve stems are thinner than predecessor TC88's. Harley also replaces inner and outer valve springs in both the XL and the Twin Cam engines with the ovate wire single spring that progressively winds in a beehive shape that tapers towards the top. They are not interchangeable with earlier models for a couple of reasons, the most important of which is the utilization of a thinner stem valve. The lighter valve correspondingly lightens the valve train, which in turn allows

for weaker seat pressures that add efficiency and parts longevity.

Now I had to fix R'dR's engine so it would still be fast but reliable.

The .570 Gear Drive Cams

I will do this by beginning with .570 lift cams run by a gear drive as opposed to the stock timing chain apparatus. This will combine with a kit that we have developed that includes some Screamin' Eagle parts. Many shops work on street engines but apply radical performance techniques that affect engine longevity.

These engines are extremely fast if built with the right combination of parts. The question becomes will it be

acceptable that longevity sacrifice for extra speed. Those with disposable income may feel that this is a tolerable trade-off.

For R'dR, he learned the hard way that the parts he chose were not coordinated but also now had to absorb the rebuild costs to make it right. Anything I discuss in this chapter on stock Twin Cam performance will be reliable. I gauge reliable performance work by having no concern if the owner rides the bike coast to coast.

I combine the S&S Cycle .570" gear drive kit with Crane 155 lbs seat-pressure-valve-springs. I also use Evolution exhaust valves in the TC88 heads, which are .030" wider diameter along with porting and polishing the stock heads.

I advise customers to use roller rocker arm kits with the .570 cams and insist on them with the .585 lift cams.

Compression

I will then plane the heads to achieve the 10.25:1 compression I want in this particular case. The reason I plane heads instead of using the 10:1 compression 1550cc pistons is that I can tailor the compression change up or down exactly to the customer's needs and or wants.

Squish Band

Only after this work do we go to the 1550cc upgrade using the H-D flat top pistons whose function is to squish the air and gas turbulently into the combustion chamber. The ascending pistons, round flattop crown has a much greater area than the smaller kidney shaped combustion chamber. The rising piston sweeps the cylinder hence the term swept area forcing the air and gas into an ever-smaller area. This compressing fuel

thoroughly mixes via the turbulence created by this squish action.

The flattop piston approaches the corresponding mating flat squish area surrounding the combustion chamber by forcing the air/gas into the chamber. This flat area on the head surface surrounding the smaller combustion chamber that meets the rising piston is a squish The Twin Cam squish band surrounds the combustion chamber.

The predecessor 1984-1999 Evolution squish band limits to one side as the combustion chamber is D-shaped. This is the single most important reason that the Evo is so much faster than the predecessor Shovel, Pan, and Knuckle engines. These previous engines have a hemispherical combustion chamber like a reverse dome. The mating piston is hemi-shaped with a dome top that fits into the chamber. No turbulence creates to

mix the fuel with resultant fat This means that the air-gas mix is too rich to burn. In other words, the ratio of gas to air is too high. This robs the engine of power since there is a lower percentage of burn. It also creates more pollution exiting the pipes. Other areas in the combustion chamber will be too lean, which is the opposite condition where there is too much air and not enough gas to burn. Some areas will mix correctly or semi-correctly for burning. The reader can discern how inefficient this hemi-system is.

Therefore, the partial Evo squish band makes the engine very much faster by mixing the gas better for a higher percentage of burn.

The Twin Cam 360-degree squish band is a step that should make the stock Twin Cam very much faster than the Evo.

## Porting and Polishing

The Twin Cam is a little faster than the Evolution, but not much. Why? The Twin Cam head flows less air through the intake port than the Evo head even though the engine is larger by 110cc. This is why porting and polishing the Twin Cam heads is so important.

The head design is very much superior to anything that went before but the ports are simply not big enough for the Twin Cam engines swept volume capacity.

## Adjustable Pushrods

Adjustable pushrods are optional when the top end is off the engine for performance work. The performance aspect of adjustable pushrods overrates in the Twin Cam but if

affordable, they sure do not hurt. The purpose of adjustable pushrods is to make the valve train stronger and lighter. However, the other reason for using adjustable rods is as a labor saver when not disassembling the top end for other purposes.

The stock Harley Twin Cam pushrods are non-adjustable. They are also strong enough because of their metallurgy and the fact that they sit almost straight up-and-down in the engine. I took the trouble to weigh the stock pushrods versus aftermarket adjustable ones and the weight differential is negligible. Sometimes the stock pushrods are even lighter.

This concept of adjustable rods being lighter mistakenly carries over from the Panhead and Shovelhead days when the stock steel pushrods were indeed heavier than the aftermarket substitutes. However, as I said if only performing a cam change for repair or hi-performance reasons, it involves extra labor to use the stock rods. More parts must remove like the

rocker box wafers on the heads in order to install the new cams.

The increasing cost of gaskets is a very real consideration when doing this job. Much labor and gasket expense reduction results by using adjustable pushrods when only doing cam work.

A long-term benefit of the adjustable rods is that as successive valve jobs occur during rebuilds over the years, the pushrods may adjust to longer positions as the valves recede into the heads because of cutting new seats.

The same applies with exotic very high lift cams, which in effect require shorter pushrods. It is a different story when installing the 1550cc big bore barrels kit by itself or with camshaft changes at the same time. The rocker cover wafers

and heads must come off in order to replace or bore out the stock barrels. In this case, there are no labor or gasket parts savings obtained by purchasing the adjustable pushrods.

However, the feel good factor may come into play knowing that better quality pushrods are in your engine even though the stock ones are likely adequate. If using aftermarket adjustable pushrods in your Twin Cam, purchase ones that come with modified pushrod tunnels like S&S or Screamin' Eagle. Pushrod kits marketed by companies like Crane do not come with the tunnels. They are very difficult to adjust with the stock pushrod tunnels as they will not collapse enough to get a good grip with wrenches when adjusting their length and torquing down the lock nut when finished. The modified tunnels supplied with S&S and S.E. collapse more fully in order to provide easy and full access to adjusting/lock nuts.

Hydraulic Lifter Limiter Kits

Before installing the pushrods, it is very advisable in most cases to add limiting kits made by companies like S&S Cycle, Inc. to the hydraulic lifters that the lower pushrod balls ride in. Four limiting rings fit inside each of the four hydraulic lifters to prevent collapsing and bleed The reliable automotive type hydraulic lifters utilized in both the Evolutions and Twin Cams are very much superior to anything that has gone before. However high lift cams and high rpm riding can take their toll by causing the lifter to bleed down causing inefficiencies and annoying noise.

The inefficiencies can translate into altered cam timing, which robs horsepower.

Limiting kits, limit the travel of the hydraulic units, which makes them, act like solids while under duress and hydraulics

during normal operation with minimal noise. They also assist in starting with high compression engines. The stock hydraulics, minus limiting kits, will bleed down while at rest. This increases the cranking compression when attempting to start the engine, which of course takes its toll on starter motors.

I have found some cam kits do not take well to limiter kits like my .650" lift Wood cams in my present ride. There are particular adjustments for special cases like this. Please see Chapter II: Repairing the Achilles sections, Hydraulic Lifter Limiter and Adjusting Performance Pushrods.

Horsepower Comparisons

Before I go any further into air cleaners and gear driven cams let us look at horsepower comparisons, which in my

opinion are usually unfair and misleading. Neither I nor the other American Iron Magazine staff feels that it is fair to compare any performance parts for a myriad of reasons. It is easy for any accomplished wrench to make it seem as if one part is superior to another manufacturer's part in terms of performance. I am not saying that all comparisons are illegitimate or deliberately falsified since that is simply not the case. However, the bad comparisons taint the good ones. The use of support parts may give unfair advantage to one featured part or another.

Even the sequence of Dyno runs will affect horsepower ratings.

Leave comparing a particular parts ability to perform to experts that have no vested interests who understand how they work especially in conjunction with support parts.

Many use the Screamin' Eagle (www.harley-davidson.com), air breather kit for both carburetor and EFI models. As I have discussed before, I could just get a K&N air filter and modify the stock breather backing plate but most riders already have the S.E. kit so why replace it with something else. At least this is the way it was.

Newer performance air-cleaner backing plates use sophisticated air horns built into them around the induction throat to transform slower moving turbulent air into smoother higher velocity laminar flow.

In the early days of the Twin Cam, this H-D performance air cleaner kit was up there in terms of performance and was one of Screamin' Eagle's best parts. The new Screamin' Eagle

air cleaner kits are still good and improve over past ones. However, in my opinion there are better ones on the market now.

Customers may want to use other breathers like Kuryakin's (www.kuryakin.com), Hypercharger with a ram flow effect or the new S&S (www.sscycle.com), breather with an enlarged pleated paper hi-flow filter, which is cool as they also work very well. S&S Cycle, Arlen Ness (www.arlenness.com), and my current favorite Doherty Power PACC air cleaners (www.dohertymachine.com), as well as many others perform wonderfully relative to the not-so-distant past. I discuss air cleaners extensively in Chapter IX: Fuel Delivery, Air Filtration and Power, and Velocity Stacks, and Cleaning Air Filters, and Performance Filters and Air Breathers, and Ram Flow Induction Systems.

Manifolds

As I previously discussed, I will use the very superior Screamin' Eagle intake tract manifold with fuel injection. In my opinion, I will not spend my money with the equivalent manifold designed for carbureted models, as I do not see much benefit.

## Exhaust

The customer decides on an exhaust system since aesthetics and sound are very important to most Harley riders. I try to give a range of choices that also deliver optimum power for the kit. Rest assured that the too fat, too long and the too loud ones are poor choices.

Collector exhaust systems are at the top of the pile particularly those with baffle adjustability like SuperTrapp (www.supertrapp.com), and Hooker (www.holley.com).

Staggered dual mufflers with a large capacity for air like those by S&S and the newer Screamin' Eagle mufflers are good choices for those that dislike collector systems.

I discuss exhaust systems at length in Chapter VIII: Eliminating the Performance Obstructions, sections, The Exhaust System, and Power Equals Breathing, and The Catalytic Converter, and Exhaust Pulses, and Mufflers Silencing Noise, and Backpressure, and Exhaust Pipe Sizing, and Exhaust Effect on Fuel Delivery, and Cam Overlap, Torque, Cylinder Fill, and Exhaust.

The Power Train

The power delivery mechanism from the engine back to the rear tire must be in perfect tune. By this, I mean that chains and belts must adjust true and with correct tension, wheels must align, engine and transmission must align, and all parts are in serviceable condition.

Many riders and mechanics screw up on the last thing we need to do, tune the motorcycle chassis and suspension. Power can be lost or regained simply by ensuring proper tire pressure, alignment and adjustment of belts and chains, replacing worn out shocks, wheel bearings and rear fork rubbers or bearings as the case may be along with wheel and fork alignments.

Adjust the clutch correctly after installing a performance clutch whether it is grabbier fibers with a stronger diaphragm

clutch spring or complete replacement with packages like the Scorpion (www.barnettclutches.com). I extensively cover clutches in the previous Chapter X: The Power Train, in section, Performance Clutching.

## The Engine Rebuild

Of course, all related internal engine bearings and bushings are in need of replacement even if they appear to be serviceable. I leave nothing to chance with either normal rebuilds or performance work.

### 1550 EFI Engine Using .585 Lift Cams and HTCC Heads

The preceding kit, I put together on a 1550 carbureted engine utilized a 10:25:1 compression ratio, .570 gear drive cams, ported and polished TC88 heads and a Crane HI-4TC ignition.

This time, I will be doing a .585 cam lift gear-driven Twin Cam engine with 95” big bore piston kit utilizing Screamin’ Eagle HTCC pistons and heads raising compression above the time honored 10:1 in a very safe and reliable manner. I will also use a Terminal Velocity ignition and gas modifier with a collector exhaust system on a Dyna and the S&S Cycle, Inc. oval performance touring mufflers on a Bagger.

Coordinated support parts, as always, makes the fastest and most durable engine. I did this engine plus the installation of a 6-speed transmission for my friend Clive on his 2004 Roadglide. Clive is very sensitive to the nuances of his machine.

Roadglide Wheelie

He commented that, “It runs really nice, good power below 3000 rpm, great power above 3000 rpm. On the highway, I dusted an Infinity G35 coupe, which I thought was good for a 1000 lb bike and rider.

“The weak link at that speed is the chassis.

“The bike didn’t feel like it was going to fall apart but it was weaving a little at over 100 mph so I didn’t push it any further even though it was still accelerating. It was fun but I should upgrade the suspension next.

”In the city I was cruising in second gear and I accelerated off, clicked her into third gear, and dropped the clutch. The result was lifting the front wheel.”

A stock Roadglide does not pull wheelies by accident in second gear never mind third. However, I should state the

Clive could really ride.

## The Powerband

The Twin Cam can rev higher than previous models because of decreased valve angles, less pushrod angularity, a shorter stroke of 4.000” and a lighter valve train. Late 2004 and later Twin Cams have an even lighter valve train due to thinner valve stems than previous models. A lighter valve train decreases internal frictional loss of horsepower and provides an ability to rev higher.

All of this is extremely important when looking at a power curve graph. The thinking mechanic chooses parts that coordinate with the exhaust. The goal is to place the peak power curve within an rpm range that the engine is capable of and more specifically an area that the rider wants to ride in.

This is generally between 2000 and 4000 rpm on a street Harley.

A knowledgeable mechanic further fine-tunes the powerband by the type of Harley, whether touring or sport. A Bagger wants the torque coming on at a lower rpm because of the mass and weight of this particular model. A 1200S Sportster may want the power curve higher as it winds through the gears because of its lightness has much more get up and go in lower rpms already.

I am going to look further at powerbands, exhaust systems, and fuel delivery. Riders ask all the time why not have the power curve really coming on at lower rpms for hole shots.

- Firstly, the powerband will begin to decrease too early for functional street riding.

- Secondly, is that Twin Cam engine cylinder efficiency is close to a 100% fill rate at lower rpms after appropriate headwork and cam-work.

- In addition, it is tough to get it that low by engine work alone although changing gear ratios is a great way to do this.

This is a time-honored way of winning short street drag races. However, it may be more practical to take advantage of gear ratios for good all-round riding or for long hauling, and less vibration with high-speed highway riding.

Gear Ratios

Gear ratios can lower for decreasing the powerband outside of the transmission by using a smaller engine sprocket with fewer teeth, a smaller transmission pulley with less nubs

or a final pulley with more nubs. The clutch sprocket cannot economically be increased or decreased in number of teeth so it becomes a neutral player in the gear ratio game.

The venerable Andrews Products pioneered the manufacturing and usage of different ratio transmission pulleys. Gear ratios can rise for a better highway powerband or to take advantage of increased horsepower and torque by doing the opposite changes, increasing the engine sprocket, increasing the transmission pulley and/or decreasing the final wheel pulley.

Going above 25-teeth on the pre-2006 engine sprocket is not really an option because of space limitations. Decreasing the engine sprocket to 24-teeth may accompany by using a shorter primary chain although this is not very necessary. BAKER Drivetrain (www.bakerdrivetrain.com) sells smaller compensator sprockets. I use a 23 or 22-tooth compensator sprocket when I want a bike to have a good holeshot.

Changing the transmission pulley by a few nubs up or down will not necessitate using a shorter or longer belt, which is good because of parts and labor expense although it is still onerous, as the complete removal of the primary system is necessary first.

There are three readily available sized early wide-belt rear pulleys, 70, 65, and 61 tooth or nubs. Any change here requires changing to a new length belt and as I said before adds considerable expense.

Rear Belt Drives

Harley and the aftermarket have both been narrowing the width of rear drive belts. This is primarily for space considerations thereby allowing fatter rear tires, which we all

know was the customizing craze. On ground-up builds, we always use the widest belt for extra strength and durability. This is true for the Aramid fiber Kevlar belts but the newest and thinnest belts utilize even stronger carbon fiber.

On Clive's .585-gear drive conversion, I lowered the engine sprocket to 24-teeth from 25; left the stock-ratio-transmission-pulley and increased his rear wheel pulley from 65-nubs to seventy.

Six-speed Transmissions

I did this in combination with a CCI Rev Tech (www.customchrome.com), .089-overdrive 6-speed transmission. The other major player at the time of this build in the 6-speed world was BAKER who also makes a variety of strong, durable, and excellent American made products. The BAKER transmissions feature full width internal gears. The

Rev Tech gears are narrower than stock width gears but are made of a stronger and more expensive material called chromemoly.

Cams and Exhaust

Low speed torque ably assists by smaller diameter 1 ¾" header pipes with correct backpressure mufflers while bigger exhaust headers will move the torque powerband further up the rpm range.

There is no way for huge power increases at low rpms without using a forced induction system such as supercharging. Nitrous will also get the bike happening more immediately. I will deal with these three performance options in a subsequent chapter, Chapter XIII: Forced Induction, sections, Ram Air and and Turbocharging, and Nitrous

Low-end torque increases can also attain with higher lift, lower duration cams, higher compression ratios, ignition timing and of course carburetor/EFI tuning.

As rpm increases, it is easy for a knowledgeable wrench to choose the right cams and exhaust to promote momentum in the flow of gases. Known in mechanical circles as scavenging, this law of physics gets exhaust gases exiting faster out the exhaust port and into the header pipes. The fast exit has the further benefit of sucking in the intake air/gas at a faster and more concentrated rate so that cylinder fills of 130% are achievable at higher rpms. Scavenging does not and cannot occur at low rpms. Longer duration cams that keep the valves open longer with more overlap where both intake and exhaust valves are open at the same time really promote the principle of scavenging the more rpms increase.

The appropriate aftermarket exhaust will allow all of this to happen via flow rates through the physics of Kinetic Wave Theory whereby the exiting exhaust gas really goes back-and-forth, pulsing in waves up and down the exhaust pipe. Exhaust pipe length, diameter and muffler volume really deals with kinetic pulses in a productive way. The wrong length with no or inadequate muffler capacity and the kinetic returning pulses are going to re-enter the combustion chamber from the exhaust pipe and pollute the incoming virgin gas. Heat and pressure differentials that occur as the exhaust gases move down the pipe are a major cause of kinetic pulsing. If an exhaust valve is opening and closing with exiting exhaust gases 3000 times a minute, there could easily be 12,000, 18,000, or more pulses traveling back and forth through the exhaust. Each pulse returning up the pipe may hit a closed exhaust valve and bounce harmlessly back out the exhaust only to return again and again until all the energy dissipates. The mechanic wants the pulse to bounce off a closed valve in order to prevent polluting the incoming gas.

Exhaust pipe length, inner diameter and of course, the muffling device if any, assists greatly in controlling and directing exhaust pulses Ideal exhaust pipe length depends on many things. This will include the size of engine, internal engine design and engine modifications. The inner diameter of the pipe and the type of muffler also dictate length.

In general, Collector 2-into-1 exhaust systems provide the best power. Faddist overlong, too big diameter straight pipes detract from power gains.

For those that are running Baggers like Clive that do not want a collector exhaust, I recommend the S&S Cycle, Inc big volume oval mufflers. There is a principle for a muffler to be truly effective at removing gases as quickly as possible that it should have 10-times the volume as the swept volume of the

engine cylinder it serves. The spent gasses should then exit the muffler through a short small restrictor pipe usually the same diameter as the exhaust header pipe. This primitive aesthetic muffler design is evident on the Japanese super bikes that look much like a tin can. S&S accomplishes this with their oval performance mufflers that also are good looking. The restrictor pipe, for example is not even noticeable unless you look for it.

Performance Heads

I will now delve into performance heads for the .585 cam lift hotrod. Screamin' Eagle has much to offer in the way of performance heads and pistons. Screamin' Eagle has 9.6:1 higher compression piston kits for the 1450 stock bore using stock heads and 10.5:1 pistons when utilizing Screamin' Eagle heads not to be confused with HTCC heads, which are for 1550cc engines. The H-D big bore kits have pistons that are 9.4:1, 9.6:1 9.8:1, 10:1, 10.25:1 and 10.5:1. Now, this sounds

like tremendous variety but they spread out to cover a variety of different head combinations.

H-D is changing their performance head selection almost yearly. I keep all the Screamin' Eagle catalogues for each year so I can access part numbers that obsolete. Strangely, I find that many of these obsolete heads are still in stock.

EFI Ignitions

The greatly improved single fire graduated spark map ignition introduced on the new Twin Cam models has difficulty coping with the lean air/gas mixtures required to meet ever-increasing emissions controls when increasing compression. Pinging or detonation is the great engine destroyer. The lean gas mixtures will have trouble with this as compression ratios bump up. The Twin Cam engine runs

better, more efficiently and faster with a 10:1 or even 10.25:1 compression ratio if adding an aftermarket ignition.

Going to 10.5:1 pistons might be asking for trouble in an Evo engine but the Twin Cam handles it well with an appropriate gas and ignition timing modifier such as a Power Commander III or another current favourite, the user friendly Terminal Velocity. Terminal Velocity incorporates a 02 sensor into your exhaust pipe to analyze the emissions. It is self-tuning continuously. Install it and just start riding. I delve into the EFI gas and ignition modifiers in Chapter IX: Fuel sections, Performance EFI Ignition and Fuel and Horsepower and Torque Bike-to-Bike and Modifying the Electronic Control and EFI Chips, Downloads, and Piggyback EFI and Output EFI and Engine Management and Power and Screamin' Eagle Race Tuner and Direct and EFI Modifier Technology Changes and Changing EFI Fuel Delivery without a and Oxygen Sensors.

## Competent Mechanics

When increasing displacement to 1550cc, H-D sells new cylinders fitted to the 3 ⅞” bore pistons whether they are stock compression of 9.4:1 or increase to 9.6:1, 10.25:1, or 10.5:1 depending on kit and year. It is very safe to do this further increase if using a competent mechanic that knows how an engine works as opposed to just bolting on parts.

Harley trained PHD technicians generally really know their stuff…especially the subjects taught at PHD University. However, many need outside training and experience to do the moves we are talking about in this performance book. The same applies about those jurisdictions that have apprenticeship programs including schooling for mechanic licensing.

Performance goes beyond knowing how to work on and fix problems on stock motorcycles although; the same principles apply to both.

## Confusing Performance Catalogues

As I have already said, Screamin' Eagle has a variety of heads available. The way they list in the catalogue is confusing to say the least. I will say that listing the parameters of performance components is not an easy task and this is not the only catalogue I must study thoroughly. In the end, I usually get to see and touch the various components over time to evaluate them better for my needs.

Having taken advanced physics combining with 35-years of performance experience and it still took me 20 minutes to sort them out in tandem with compression changes and piston

dome design. It is important to choose well in order to get the best bang for your buck.

With our .585 cams lift engines, I go for the gusto and choose the HTCC heads combining with the higher compression HTCC 1550cc pistons. These pistons made by Arias are works of art. They match to the heads perfectly. Deep valve pockets machine into the piston crowns to accommodate valve lifts up to .650".

They advertise an rpm limit of 7000 rpm.

Limitations of Pushrod-operated Engines

I do not go to these ultra high cam lifts on a customer street bike, as pushrod-operated engines are limited mechanically. Well, I do on my own bikes just for fun… sometimes, expensive fun.

Overhead cams that directly actuate the valves can handle these lifts but poor old dinosaur technology pushrods just cannot move fast enough to handle them in a street reliable way. These figures begin to violate the laws of physics on a pushrod-operated engine although the Twin Cam is the one engine of all the Big Twins that is most capable.

S&S makes a 124 cubic inch Twin Cam designed that have .640” lift cams. They use superior metallurgy: lighter parts and performance design configurations that help this engine stay together. S&S remanufactures the whole engine to do this. There is not one stock part on it. Please do not install a cam that tall or rev your bike to 7000 rpm on a pushrod operated street bike unless you are wealthy and have easy access to tow trucks.

I must confess that I will use .650” lift cams in my own Twin Cams and surprisingly, the engines hold up well. I do this, knowing that I may pay the price one day.

HTCC Arias Pistons

The intricate shape of the HTCC Arias piston dome is all business in promoting turbulence of air and gas so that it can approach the highest percentage of burn. The flame travel movement once ignition occurs is orchestrated beautifully to every part of the combustion chamber to approach 100% burn of the increased air and gas brought in by the bigger port heads.

The HTCC head intake valve increases in size to 2.000” while the exhaust increases to 1.623”.

S&S Performance Heads

S&S also have pistons that combine with their heads and are similar works of art with a different design. The S&S Twin Cam heads come in two basic sizes with 79cc and 89cc combustion chambers and different size valves.

The 79cc heads are for 88”, 95”, and 100” engines producing compression ratios of between 9:1 and 10.3:1. The 89cc heads are for the engines up to and beyond 116 cubic inches. Their mega 4 ⅛” bore engine use special 91cc combustion chamber heads, as this bore would increase compression too much if using an 89cc combustion chamber. Please see Chapter VIII: Eliminating the Performance Obstructions, sections, Edelbrock Performer RPM and Screamin’ Eagle Performance and Super Stock Performance Heads.

On carbureted models I use the S&S Super E on the .570 lifts and go to the “G” on .585 and larger. Many mechanics prefer the “G” on the .570 lifts but I think the venturi is a little large at this stage.

I prefer the 45 Mikuni Smoothbore for medium sized strokers until the venerable S&S Super G or Super D takes over once more. Of course, there is now a 48 Mikuni Smoothbore marketed separately or in a kit by Rivera Engineering (www.riveraengineering.com). Please see Chapter IX: Fuel Delivery, sections, Carburetion, and Ignitions Coordinating with and Tuning the CVH Carburetor, and Tuning the S&S Super E and G and Tuning the Mikuni Smoothbore Carburetor.

However, for the knowledgeable, EFI will win the race

every time and with less gas!

## The Best Engine

This Twin Cam engine, outside of weak timing case components, is the best the Factory has ever built excepting the V-Rod. It is the most precise. The Twin Cam is with excellent metallurgy. It is the best design Big Twin and it is the strongest of them all. It is a superb platform from which to extract reliable and big performance gains.

## The Bottlenecks to Power

As I discuss many times there are three bottlenecks to achieving good power with the Twin Cam engine.

The first one of course, is insufficient flow through the heads. I deal with this impediment by installing the HTCC heads on our .585 cam engine.

The second impediment is the weak timing chain support system that solves by installing an Andrews (www.andrewsproducts.com), Crane (www.cranecams.com), S&S (www.sscycle.com), or Wood (www.woodcarbs.com), gear drive cam system that allows the usage of stronger harmonically wound performance valve springs and corrects any wandering cam timing. Please see Chapter II: Repairing the Achilles Heel, sections, The S&S and Knight Prowler Silent Belt Drive and The Advantage Summary of Gear Drives over Timing and Gear Drive Installation Instructions and Knight Prowler Belt Drive Installation: 1999 to Present. Also, view Chapter IV: Cams, Cams, Cams, sections, Cam Lift and Valve and Potential High Lift Cam Lobe and Cam and Valve and Choosing a

The final bottleneck is the restrictive emissions controls. I deal with these concerns by modifying the ignition/fuel maps, a free-flowing Air Breather, longer duration cams with more overlap and a non-restrictive exhaust.

## The Break-in

In the last major heat wave in Toronto where temperatures were, approaching 100 degrees F we modified a fellows Road King with oodles of performance components using the basic .570 cams lift kit described earlier. He had come from Texas and thought the temperatures were relatively mild. I had no idea he was going on a long trip with a loaded Road King and trailer immediately after our work in the blistering heat. I panicked. I am overprotective and encourage customers to break in their bikes without a load or a passenger in cooler weather or at least at night. He wrote me a month later to say

that he had 4000 miles on his bike since the rebuild and that the engine was working perfectly.

Our friend Philip wants his Dyna to look the way he wants it to look and he wants it fast the way he wants it fast. Do what he wants in the best quality there is in a quick efficient manner and he will not blow you off. Therefore, we do the .585 performance build for Philip, which is cool, except we are in another heat wave of 95 degrees, day after day. He shows up his girl on her Sportster to pick up the Dyna and starts strapping his baggage on. I say can you break the bike in late at night. He responds that he is leaving to ride around the Great Lakes with a bunch of side trips and he will break in the engine on the way but he cannot do it at night. Well! The Great Lakes on the Canadian side involve at least 1700 miles and who knows how much on the U.S. side. I am no longer having a good day. A couple of weeks later Phil rides up to the shop after completing his trip with the engine running like

a fine Swiss watch. He says, “Now that’s a fast bike!”

The Twin Cam is a wonderful bike to work on. An Evolution may or may not survive the above two scenarios whereas the Shovel would be toast. The more I work on Twin Cams, the more I must shed preconceptions that dog my brain that center on the foibles of previous Harley-Davidson technology. Please see Chapter III: H-D 110 Cubic Inch CVO, section, Break-in and Chapter XII: Cubic Inches: Strokers and Big Bore, section, The Break-in Procedure.

# Chapter XII

## Cubic Inches: Strokers and Big Bore

There is no other engine produced in the history of Harley-Davidson that is more amenable to huge performance gains via increasing the stroke of the flywheels in combination with bigger bore pistons. The Twin Cam serves as a reliable and strong platform for major modifications that would destroy earlier engines in short order. The engine weighs in at 165 lbs, which is heavier than previous models, but the extra weight derives from quality. The bore of both the TC88 and the TC96 is 3 ¾" a hefty increase over the Evolution's 3 ½" while the TC88 stroke decreases to 4.000" from the Evo 4 ¼". However, the stroke increase to 4.375" that results in the

extra cubic inches for the TC96 is .125” greater than the Evolution, which is again, 4.250”. The TC88’s lower stroke means more reliability with slower piston speed and allowing for the potential of slightly higher revs. I know the reader has read the foregoing earlier in the book but this information provides the basis for what is to come in this chapter.

## Defining Stroke

Stroke defines as the distance the piston travels up the cylinder from BDC (bottom dead center), to TDC (top dead center). The volume of air displaced by the piston on its stroke is the swept The cubic inch (c.i), or metric cubic centimeter (cc), of an engine measure the swept volume that defines as the volume or displacement of the engine.

H-D de-stroked the TC88 Twin Cam engine from that of

the longer stroke Evolution engine. Stroking is short form for increasing the piston stroke and involves using flywheels that place the crank pin closer to the outer edge. As the flywheels rotate, the connecting rods guided by the crank pin, push the pistons up further and then pull them down more distance than on a stock stroke bike. Since the stock TC88 Twin Cam's stroke is a ¼" less than that of its predecessor Evo, the crank pin is moved in a little closer to the centre of the rotating flywheels. Since this is a diameter circular motion, the distance would be a radius of ⅛" closer to the center for a ¼" decrease.

The moving of crank pin location is not a mere adjustment. Different flywheels must be used with the crank pin mounting holes bored in a different location, usually closer to the rim of the flywheel since most times we want to increase the stroke resulting in greater swept volume also known as engine displacement.

Sometimes, especially in the old days, we would de-stroke an engine for faster revving capabilities for specific racing applications but this is almost unheard of for street racing. In times past, there were fewer options for making an H-D engine faster in terms of parts available. Furthermore, the quality of the older engines made it difficult for them to have longevity before a rebuild when performancing them.

What makes performancing a Twin Cam so exciting is the array of quality parts availability that gives a multitude of safe options for going faster.

## A Change in Stroke Leads to Other Modifications

I will also describe at length the platform the Twin Cam engine provides for dramatic and safe increases in displacement through stroking and bigger cylinder bores.

The fact that increasing the stroke means that the pistons will push higher in the cylinder to a new TDC and be pulled lower to establish a new BDC normally requires longer barrels so the pistons don't smash into the heads at TDC.

Another technique to prevent the piston skirts from smashing into the flywheels at BDC is to use smaller diameter flywheels. Obtaining smaller diameter flywheels is not an issue as they are readily available in combination with increased strokes.

The usage of longer cylinders makes the engine taller, which in extreme cases can make fitment of the engine in the frame an issue. This also makes it necessary for wider manifolds, as the distance between heads will also increase. Longer pushrods may also be necessary.

Internally, it is essential to check many tolerances and to perform some modifications. The crank pins on Evo and older models have crank pin nuts holding the flywheels and crank pin firmly in place. Since, the diameter of rotation of the crank pin nuts increases inside the engine casings, many times the machinist must remove material from inside the casings to allow this travel.

Twin Cam crank pins press into the flywheels flush with their outer surface. Therefore, they do not protrude beyond the side surface of the flywheels so rotational interference with the inside casings is not an issue. However, the increase in stroke means that the male-female fitted connecting rods have an increase in rotational angle on all V-Twins with angularity. As the con rods rotate, particularly through the 9-

o’clock and 3-o’clock positions, the inside bottom shanks of the con rods near the con rod big ends that house the crank pin and crank bearings will interfere by coming in contact with each other. The performance mechanic must check this area for sufficient tolerance. If there is not enough rotational room, the mechanic must clearance this area to obtain a sufficient tolerance clearance of .060”.

## Boring Crankcases for Bigger Cylinders

Boring crankcases for larger bore barrels will often mean that the centre case bolt between where the cylinders sit in the engine casing spigots that clamps the casing halves together must similarly machine for clearancing. The increase in spigot hole sizes will sometimes run into the sides of this center bolt. If this occurs, the bolt must install with cutouts facing their respective cylinder. If not, a perfectly round part

of the bolt will sit protruding into the spigot holes against the cylinders; pressing against them, distorting their inner concentric integrity with severe consequences to the pistons. One can see that the rule of thumb in the world of strokers is that one change leads to another.

## Stroker Pistons

A common way to deal with the stroker travel increase of the pistons that may or may not be in conjunction with longer cylinders and/or smaller diameter flywheels is to use pistons that are shorter than stock ones. This use of a shorter piston is normally in conjunction with the relocation of the wrist pin hole for the same reason of avoiding collisions. The pistons will not have the same longevity as stock pistons because there is not as much mating supporting surface with the cylinder (barrel) walls. Using superior metallurgy partially deals with this wear issue. This is not a big deal with milder stroke increases up to 4 ⅝" although the shorter the stroke the

more reliability that ensues. However, it still is a problem in terms of premature wear with radical increases up to 5 ¼". Strokes of 4 ¾" will decrease engine longevity noticeably but 5.000" is meant for run passes at the drag strip where frequent rebuilds are the order of the day. Of course, it only follows that 5 ¼" strokes are the true engine killers but they sure go fast in the meantime.

Engine Killers

The two engine killers are piston speed and consequential heat. The shorter piston must travel both higher at the top of the stroke and lower at the bottom of the stroke as the flywheels rotate. The piston must traverse this extra distance in the same time as that of an engine with a stock stroke. The only way to do this is by increasing the speed of the piston. Increase in piston speed leads to a corresponding increase in

heat. Suffice it to say, there is a safe range of piston speeds within which the bike can operate.

Moreover, the more exaggerated the stroke, the more likely piston speeds will approach the limits of their physical properties.

## Safe Strokes

Modern strokers use superb quality materials in conjunction with the best stroker platform in H-D history, the Twin Cam engine. This allows the safer usage of increased piston speed within reason. Any stroker combos mentioned in this chapter will be safe. If otherwise, I will clearly note it but as I have already said up to 4 ⅝" stroke is safe in the Twin Cam platform especially when broken in properly, warmed up before use, oil kept clean and avoiding high rpms unless necessitated by the throes of a street race.

# Piston Speed

Piston speed is the rate at which the piston travels up and down the cylinder, measuring in feet per minute (fpm). At higher rpm, the piston travels from 0 mph at TDC and BDC to over 100 mph and back to 0 during each of the four strokes completing one engine cycle of engine crankshaft rotation. The longer the stroke the faster the piston has to travel to cover the distance of its stroke during the engine's revolution of rotation. Piston speed is very important because there are mechanical limitations to how fast a piston can travel, slow down, stop, reverse axis, speed up, and keep repeating this thousands of times per minute. If piston speed exceeds the structural strength limitations of the connecting rods and pistons then extreme stress and possible failure will ensue.

Strokers increase piston speed exponentially for every increment in stroke length. Qualities of mechanical parts like the Twin Cam pistons, which are hypereutectic and thus superior to Evo eutectic pistons and earlier Shovelhead hypoeutectic construction allows for increases in average piston speed. See Volume I: The Twin Cam, Chapter I, March 1998: Introduction of the Twin Section, Dissecting the Twin Cam 88 and subsection, Hypo, Hyper and Eutectic

## Internal Engine Parts Speed Limitations

The amount of power a Harley-Davidson engine generates relates directly to the mechanical limitations of their internal parts speed. The safe speed of various internal components will limit by metallurgical strength. Valves, pistons, rings, and connecting rods suffer severe forces from acceleration rates that determine by engine rpm to the point where physical breakage and piston ring flutter can occur with resultant power loss and possible engine destruction.

Piston Ring Flutter

Piston ring flutter occurs when combustion chamber pressure leaks down past the top compression ring (blowby), but does not get by the second or middle ring. Many feel that this creates pressure between the top and middle ring that pushes down on the second ring and up on the top ring. This pressure lifts the top ring off its ring land creating ring flutter. There are a couple of ways other than piston speed reduction to reduce flutter such as opening the gap on the second ring thus allowing the pressure to escape downwards towards the oil scraper rings. This invites further blowby, reducing the ring's sealing capability resulting in cylinder seal and power loss.

If an engine spins too quickly, valves may float in the

combustion chamber and/or bounce off their mating valve seats. Power loss is the least that will occur but if a valve contacts the piston crown, then the power loss will become permanent, necessitating a rebuild.

The connecting rod to stroke ratio that I discuss in Chapter VII: Torque and section, Connecting Rod Ratios describes the relationship between the length of the con rod to the length of the engine or more properly flywheel assembly's stroke.

## Performance Connecting Rod to Stroke Ratios

Increasing the connecting rod to stroke ratio will result in a lower piston speed. The engine builder can achieve a taller ratio by using a longer connecting rod or a shorter crankshaft stroke or a combination of both. This longer con rod to stroke ratio can potentially create more power. The longer

connecting rod delivers more force from the piston as it departs tangentially to the crankshaft's rotation thus delivering an increase in torque.

A shorter con rod to stroke ratio will create higher piston speeds, but this also may be beneficial. Increasing piston speeds can develop extra turbulence in the combustion chamber and reduce detonation. Increased piston speeds also draw fuel-air mixture into the cylinder faster via scavenging especially if the induction diameter increases boosting cylinder fill beyond one hundred percent.

Changing the Con Rod Ratio

Connecting rod length and stroke are independent and fixed although they coordinate. Obviously, different con rod lengths will alter the ratio but how can this be done and have the engine still function? Changing the cylinder-spigot engine-

case deck height is one way, albeit a complicated one. An easier and more practical method is to use pistons with a raised or lowered wrist pin location.

## Tall Con Rod Ratio

As I have said before, a connecting rod that is longer in relation to a particular stroke causes the piston to stay a longer time at TDC. Furthermore, the piston will move toward TDC and away from TDC more slowly than occurs with a shorter connecting rod. Tall con-rod-ratio engines also increases suction above the piston with less force because the piston moves away from TDC more slowly. Therefore, longer connecting rod engines usually produce lower intake-port air velocity, which will reduce low-speed torque. However, longer con rods, also place less thrust load on the cylinder walls. This reduces wear by generating less parasitic drag

resulting in less frictional power loss, particularly at higher Moreover, longer con rod ratios generally produce more power due to reduced engine drag reduction especially at higher rpms. A longer con rod relative to a given stroke generates less stress on the component parts because of the lower rate of piston-speed acceleration approaching TDC and moving away from TDC.

The average piston speed of a long-con-rod engine and a shorter-con-rod motor is the same. The difference is that the peak piston speed is less with longer connecting rods. Therefore, the rate of piston speed changes relative to the flywheel assembly depending on connecting rod length.

Taller con rod ratios approaching 2:1 will increase power in the upper rpm ranges, thus sacrificing low-speed torque.

Piston wear will decrease.

## Shorter Con Rod Ratio

Shorter con rod engines have opposite characteristics. They apply more force to the crank pin as crank to-rod-angularity increases above after bottom-dead-center. Short con-rod-ratio engines will develop more torque at lower engine speeds with an earlier end to the power band in terms of higher rpms. Lower con rod ratios of say, 1.45:1 produce torquey engines with low rpm power bands that will run out of breath as higher rpms approach. Piston wear will also increase. The TC96 has a shorter con rod ratio than the TC88.

## Medium Con Rod Ratio

In general, medium connecting rod ratios (e.g. 1.6:1) produce engine reliability and longevity.

# Measuring Average or Mean Piston Speed

The reader can calculate average piston speed (aps) in feet-per-minute (fpm) by multiplying the engine stroke times the rpm and then dividing by 6, which is a constant. If studying the following formula, it will become readily apparent how the constant of 6 evolves, which converts inches to feet.

Average piston speed is also known as mean piston speed (mps). Average or mean piston speed is a rule-of-thumb measurement in engine evaluations. It calculates by showing the average velocity of a piston at a given rpm in an engine with a known stroke length. The piston travels a distance equal to twice the stroke length every revolution. Therefore the formula is,

mps (ft per minute) = rpm x 2 x stroke (inches) ÷ 12 (inches

per foot)

This formula condenses to,

$$mps = rpm \times 2 \times stroke \div 12$$

This formula condenses to,

$$mps = rpm \times stroke \div 6$$

Calculating Average Piston Speed from Stroke and Rpm

fpm = feet per minute

Average piston speed (aps) calculates in fpm.

Stroke measures in inches.

The figure 6 is a constant

Formula: aps/mps = (rpm x stroke) ÷ 6

Example

Example; TC96, 4.375-inch stroke at 3500 rpm

aps = (stroke x rpm) ÷ 6

aps = (4.375 x 3500) ÷ 6

aps = 15312.5 ÷ 6

average piston speed (aps) = 2552 fpm

Example; TC96, 4.375-inch stroke at 5500 rpm

aps = (4.375 x 5500) ÷ 6

aps = 24062.5 ÷ 6

average piston speed = 4010 fpm

Example; TC88, 4.000-inch stroke at 5500 rpm

aps = (4.000 x 5500) ÷ 6

aps = 22000 ÷ 6

average piston speed = 3667 fpm

TC88 and TC96 Average Piston Speeds versus RPM

*Aps = average piston speed

**Maximum aps is approximate and for comparative purposes only

Many generally accept that engines with a cast crankshaft, stock connecting rods, and cast pistons will have a sustained maximum piston speed of 3500 feet per minute (fpm). This would be applicable to say, a late Shovelhead engine with eutectic pistons and straight connecting rods.

## Evo Piston Speeds

Better engines like the Evolution that also use eutectic pistons but with better quality offset connecting rods might have a sustained piston speed of say, about 4000 fpm.

## TC88 Piston Speeds

A Twin Cam TC88 with very strong connecting rods and hypereutectic pistons will increase approximate sustained piston speeds to 4250 fpm. The reader can see from the above formula that a TC88 has an average piston speed of 3667 fpm at 5500 rpm. Therefore, the Factory has a built-in safety cushion. The TC88 has an average piston speed of 4000 fpm at 6000 rpm. The engine is getting dangerously close to extreme stress where parts may compromise. TC88 has an

average piston speed of 4334 fpm at 6500 rpm. In my opinion, the TC88 will begin to self-destruct if staying at this 6500 rpm for any length of time. The rider may want to approach this rpm if the power band is still operational but why would it be unless a drag strip engine?

TC96 Piston Speeds

A Twin Cam TC96, also with very strong but lighter connecting rods and hypereutectic pistons will increase approximate sustained piston speeds of 4400 fpm. The increase in maximum average piston speed for the TC96 is a direct result of lightening the connecting rod small ends by tapering them from bottom to top. In addition, the TC96 has a lighter piston that is not only shorter but whose non-thrust sides do not protrude as close to the cylinder walls as its TC88 counterpart. Moreover, the TC96 uses a much shorter and thus lighter wrist pin. All of the above combine to allow higher piston speeds.

From the above formula, a TC96 has an average piston speed of 4010 fpm at 5500 rpm. Therefore, once again the Factory has a built-in safety cushion. The TC96 has an average piston speed of 4375 fpm at 6000 rpm. The engine is getting dangerously close to extreme stress where parts may compromise. The TC96 has an average piston speed of 4740 fpm at 6500 rpm. The TC96 pistons will begin to self-destruct if staying at this 6500 rpm for any length of time.

Both increasing metallurgical quality and therefore, strength combining with lighter parts will increase the maximum average piston speed to approximately 5000 fpm.

Piston Speed Stress

The piston stops at TDC and BDC but accelerates to speeds

that can exceed 100 mph in between. This places unbelievable stress on these reciprocating parts. Any reduction in mass (weight) will result in alleviating some of this stress.

## Piston Acceleration Speeds

Any thinking person realizes that the piston speed in its bore is not constant since the piston must stop at BDC and TDC, speed up after stopping, and slow down again before stopping. Piston acceleration is simply a measure of how fast piston speed changes. If piston speed does not change, there is no acceleration.

Conversely, if piston speed changes rapidly, which it does then there is a lot of acceleration. The study of piston velocity and acceleration provides some interesting insights into the mechanical challenges in a reciprocating engine.

For our purposes here, I will look at the crankshaft as it moves through 360˚ of rotation, which is two strokes in a four-stroke engine that requires 720-degrees of rotation to complete one full cycle. Obviously one stroke of the four will take of crankshaft rotation. I need 360-degrees of rotation to complete two strokes to show piston velocity relative to two positions, which are top dead center (TDC), and bottom dead center (BDC of piston travel. Piston travel during the first of the rotation completing one stroke moves significantly farther than it does during the second

During the first of rotation, the piston travels approximately 60% of its stroke not the anticipated fifty percent. Piston velocity at TDC and BDC will obviously be zero. Furthermore, maximum piston velocity will occur at about before and after TDC, not at as initial logic would dictate.

Moreover, the piston velocity at any rotation point after TDC and before maximum velocity is greater than at the same number of degrees before BDC. For example, the velocity at say, of crankshaft rotation after TDC, might be about 60% maximum velocity. Compare this with the velocity at of crankshaft rotation before BDC, which might be substantially slower at about 38% of maximum velocity.

I am using educated figures as examples because maximum velocity varies directly with the engine rpm. This asymmetric velocity profile is a result of geometric characteristics. Determining the position of maximum piston velocity depends on the relationship between the connecting rod length and the stroke length.

The basic formula of stroke length (L), over connecting rod length (R), or simply L/R is the connecting rod length divided

by the engine's stroke length. As the connecting rod gets shorter with respect to stroke, two interesting consequences occur that can have a dramatic effect on cylinder fill. If the formula expressed as S/C, it would be easier to remember as S = stroke length and C = con rod length.

- First, the point of maximum piston velocity moves closer to TDC.

- Secondly, the piston moves away from TDC faster, creating a stronger intake event.

The crankshaft degrees-rotation of maximum piston speed will influence camshaft lobe profile design particularly the intake cam lobe. This allows optimization of the intake event in a particular speed range.

## Acceleration Force

The force it takes to accelerate an object is proportional to the weight of the object times the acceleration. The weight of reciprocating components becomes crucial because of the multiplication of force generation. This is another reason the TC96 found it necessary to reduce reciprocating weight by lightening the pistons, wrist pins, and connecting rod small ends. Piston acceleration is important because many of the significant forces exerting on the pistons, wristpins, connecting rods, and bottom end bearings directly relate to piston acceleration.

## Vibration: Long and Short Connecting Rods

Piston acceleration is also the main source of an engine producing external vibration. It is clear that the engine with a long connecting rod relative to stroke produces a small L over

R ratio. A small L/R ratio produces an earlier velocity peak that causes vibration.

A shorter connecting rod relative to engine stroke produces a larger L/R ratio giving a peak velocity that helps reduce vibration. I must say that each concept is fairly simple, once explained when viewing in isolation. However, when looking at them all together it does get overwhelming…for me anyway.

## Engine Maximum Rpm Calculation

I am going to use conservative figures in order to keep the avid reader out of harm's way.

If dividing a figure of 24000 by the engine stroke for the motor using very strong connecting rods and hypereutectic

pistons, the reader will get an approximate-redline rpm.

Therefore, the TC88 engine will have an approximate-redline of 6000 rpm (24,000 ÷ 4.000).

A TC96 engine will have an approximate redline of 5486 rpm (24000 ÷ 4.375). However, this formula demonstrates the heavier reciprocating parts of the TC88 will skew the results. The formula does not consider this weight factor.

## Over-Square, Square and Under-Square Engines

A square engine is one with an equal bore and stroke. For example, if I increase the bore of a TC88 engine to 4.000”, I will make a square engine since the stroke is also 4.000”

An over-square engine will have a bore diameter greater than its stroke. For example, if I increase the bore of a TC88

engine to 4 ⅛”, I will make an over-square engine since the stroke remains at 4.000”. Over-square engines achieve higher rotational speeds because the pistons do not travel as far.

An under-square engine has a bore diameter that is smaller than its stroke.

The TC88, TC95, TC96, TC103, CVO 110, Evolution 1340, Shovelhead, Panhead, and Knucklehead engines are all under-square. Harley-Davidson is famous for its torque. Under-square engines cannot rotate as quickly, but generate more torque at lower rotational speeds.

## The Power Platform

### Bore

The TC88 and TC96 Twin Cams 3 ¾" bore that is bigger than the previous Evolution and late Shovelhead 3 ½" bore and even larger than the 3 early Shovelhead bore H-D engines does not affect reliability but allows for more cubic inches. This is sort of a reverse stroker with the same torque and thus horsepower increases. This combines to allow a sustained engine speed of 5500 rpm on a stock Twin Cam, which is higher than any stock Harley that has gone before.

Compression

Compression also increases for more low-end oomph up to between 8.8:1 and 9.0:1 on the TC88 and 9.2:1 on the TC96. Previous engines go as low as 7.5:1 and as high as 8.5:1. Furthermore, we have ourselves a fast machine here that easily gets much faster but as I have already stated; it is going to be the most reliable engine ever used by the Factory.

Cubic Inches

Eighty-eight cubic inches convert into 1450cc while 96 cubic inches translates into 1584 cc.

Big bore conversions from 3 ¾" to 3 ⅞" that increase the

TC88 to 95” and the TC96 to 103 cubic inches do not affect reliability one iota. Piston speed remains the same as with the TC88 for the 95-inch kit and with the TC96 for the 103-inch kit.

I will show later on that this is the reason a Twin Cam can increase easily to 116 cubic inches or to 124 inches primarily through bigger bore increases combining with a 4 ⅝” stroke. This is very reliable way since piston speed stays within relatively safe parameters inside the realm of comfortable physical principles.

Horsepower

Horsepower on a stock TC88 is in the low sixties hovering around 62 HP at the rear It may not seem like much but a stock Evo probably has around 45 ponies at the rear wheel. Doing the math means that an extra 17-ponies where the tire

contacts the asphalt is about a 33% real increase. These are averaged real figures not the perfect mythological figures taken at the engine in laboratory conditions.

Torque

Torque is 86 ft-lbs For the TC88 FLHT models while 82 ft-lbs for the Dyna. Why is there a difference? The Bagger mufflers are bigger and can handle more volume. The ability to breathe more is the first principle of hi-performance. I discuss at length in Chapter VIII: Eliminating the Performance sections, The Exhaust System, and Performance Exhausts, and Power Equals Breathing, and The Catalytic Converter, and Exhaust Pulses, and Mufflers Silencing Noise, and Backpressure, and Exhaust Pipe Sizing, and Exhaust Effect on Fuel Delivery, and Cam Overlap, Torque, Cylinder Fill, and Exhaust that bigger volume muffler(s) is one major

component amongst others in a performance exhaust.

Comparative torque figures for the Evo FLHT are 78 ft-lbs at 100 rpm more. Of greater importance, the Twin Cam torque curve repositions into the lower rpm area producing more useable power during initial acceleration. Transient torque addresses off the line in order not to be soft so the rider is going to feel power sooner than with previous models. The TC88 heads even have an AR or anti-reversion (tork or torque valves), step built into the exhaust port to create more low-end power.

## The Ultimate Torque Arm

An external change is the ultimate torque arm. The engine and transmission move closer together so they almost appear to be uni-construction. Previous Big Twins had the primary system serving double duty housing the primary drive and

clutch as well as keeping the engine and transmission in line. This system is below par as it cannot control flexing of engine and transmission. Rigidity is another theme that pervades the Twin Cam. This translates into better handling, reliability, and durability.

## Pushrod Angularity

Twin Cam pushrod-angle-holes are almost parallel straight up and down. Evo and earlier pushrod angularity with their resultant problems largely eliminate with the side-by-side the dual cams.

## Breather Apparatus

The rocker box wafers on top of the heads hold the breather apparatus, which consists of the oil separator, the

flapper valve and the scrub sponge to control blowby also known as oil carryover. Even though this is the most efficient breather system of all Big Twins and Sportsters, some Twin Cams still have oil reaching the outside world much to the consternation of the rider.

Stroking and making the cylinder bore bigger increases the swept volume of the larger diameter pistons that travel further up and down the bore. The increase in volume of air displacing back and forth inside the engine caused by ascending and descending pistons also dramatically increases the chances of oil blowby out the air breather.

The mechanic must address blowby potential whenever stroking and boring an engine as the stock oil and air separating breather apparatus is not able to handle the increase in movement of oilified gases. I will discuss modifications like using an S&S reed valve behind the oil pump and TP Performance rocker covers to control blowby in

a following section, The Many Cures for Puking Oil.

## Piston Wrist Pins and Bushings

In continuing the Twin Cam platform for big safe performance increases, the piston wrist pin hole offsets slightly. It is big with a corresponding large wrist pin bushing pressed into the small end (top), of the connecting rod that is meaty enough for the biggest engine. An Evo wrist pin is .792” thick while the Twin Cam is a whopping .927” in

Another first for H-D are piston oil cooling jets that squirt oil up into the undersides of the pistons once reaching acceptable lubricating oil pressure. The squirting oil cools piston dome temperatures by about 50-degrees Fahrenheit.

## The Engine Casing

The engine cases are of paramount importance since they will support both the stroker flywheel assemblies and big bore cylinders that I will be using to create big cubic inch displacements. High-pressure die cast aluminum that split down the middle like always, these best of all built Big Twin engine cases promote rigidity, strength and durability as the framework for what is inside and on top. The 360-design formulated proprietary alloys are special for these cases.

The quality is evident when looking at them. No more leaks. No more cracks.

A major problem when manufacturing motor casings is how many times it is necessary to clamp them for a machining process. With every clamping, the alloy distorts which can cause machining distortions as the alloy bounces back to its former shape before clamping. Evolution cases require

clamping 32-times during their machining process. The Twin Cam cases need only 5-clampings.

The Flywheels

The quality is on a level with S&S Cycle, Inc. flywheels except everything is obviously stronger which is a mouthful as those of you familiar with S&S probably regard their components as being some of the best in the industry.

Connecting Rods

S&S stroker con rods are big but these are bigger making the Evo ones look puny. The straight, .007 interference press fit, and 3 hole oiling crank pin is so muscular that it's about a ½" thicker, (.420"), than the diameter of that relatively skinny old single oiling hole crank pin used in the Evo.

The connecting rods look the same, just bigger with more muscle everywhere. Ditto with the captured 3-sets of 18 needle rollers in copper-coated cages that help break-in procedures, durability and offer easy wear spot identification. There are three more bearings than the Evo and they are thicker in diameter and longer in length.

I have gone through all of the above information although I have left just as much out to demonstrate the clear superiority of the Twin Cam over what has gone before.

The reader can now see that the Twin Cam platform will allow bigger and more reliable big bore strokers. Why would anyone build a big inch engine utilizing stock Evo or Shovel engine casings with technology not up to par with the Twin Cam? It is the most reliable platform to date, which is a major reason for a very reliable 124-incher utilizing stock casings.

The engine configurations I am going to discuss will eliminate the three bottlenecks to achieving good power with the Twin Cam; insufficient flowing heads will replace with those capable of providing all the gas and air an engine could possibly want. I will demonstrate replacement or modification of emissions control parts like the exhaust, restrictive breathing, cams, and ECU computer fuel and spark tables to ensure big power gains. Lastly, gear drive cams will upgrade the weak timing chain apparatus.

## Big Bore Stroker Components

Strokers and/or big bore engines need special heads with bigger ports and valves to flow the air for a much greater swept volume of the cylinders. Cams must be of a higher lift and duration in order to open the valves further and for a

longer period, again to handle bigger volume air requirements. Breathing apparatus like the air cleaner and the exhaust must open up for greater flow volume capability, again of flow. The engine is like a human lung. It must be able to breathe in and out the increased volumes of air in order to produce the power we are after.

Engine Casings

S&S sells stock 3 ¾" bore engine casings but also 4.000" big bore and 4 ⅛" bigger bore casings, clearanced for strokes up to and including 5.000" from the stock 4-inches (TC88) or 4 ⅜-inches (TC96). Cam chest clearancing for stock .473" lift cams up to .675" lift cams is another necessary feature. Special order casings are available with up to a 4.380" spigot bore and different cylinder stud patterns. A nifty option with all their engine casings is a choice of rear motor mount style,

either Twin Cam or Evolution. Twin Cams bolt directly to the transmission case for providing greater rigidity. Twin Cam rear motor mounts also requires a Twin Cam style frame. The Evo rear motor mount style allows for a wider variety of frames as well as the usage of an Evo style separate transmission. Of course, I would question why any serious performance enthusiast would prefer this setup for anything but styling. Twin Cam engine mounting promotes rigidity and substantially reduces counterproductive powertrain flexing.

Big Inch Kits

The following list does not include everything available of the kits that are available for A and B motors using stock engine casings but is still indicative.

- The 100 cubic inch big bore kit achieves the time-honored square motor with a 4.000” bore combined with the

stock 4.000” stroke.

- The 106-inch kit uses H-D 3 ⅞” (95 or 103 kit) bore cylinders with a 4 ½” stroke that is very safe.

- The 116”-kit uses 4.000” bore cylinders and 4 ⅝” stroke flywheels. The stock casing cylinder spigot holes can very safely bore out to accept these oversize bore cylinders.

- The 124”-kit will use 4 ⅛” bore cylinders and 4 ⅝” stroke flywheels. Again, it is very safe to use the stock cases with this engine configuration,

The stock engine case spigot holes are approximately 4.075” in diameter. The cylinder fits into a spigot hole in the engine casings and bolts to its deck on the engine casings. The spigot must bore out to 4.225” to accommodate the 4.000”

cylinders while the 4 ⅛” cylinders require a spigot hole size of 4.322”.

This precise work is safe to do as long as the machinist is competent and the correct tooling is available for the job. A thinking person must do this work since it requires allowing center-case-bolt fitment, the avoidance of hitting oil passages and the cam-bearing boss. It is of paramount importance to anticipate the effect on engine components with each engine in performing each modification. Mechanics that do work by rote have no place in this type of engine building. The successful performance wrench must have a clear understanding of physical principles, metallurgy and mechanical principles plus their limitations. He or she must be passionate about their work.

Heads and Pistons

S&S also has pistons that match and work with its hi-performance heads, and are similar works of art with a different design.

The S&S Twin Cam heads come in two basic sizes with 79cc and 89cc combustion chambers and different size valves. There are also special order 91cc heads for the 4 ⅛” larger bore. The 79cc heads are for 88”, 95”, and 100” engines, producing compression ratios of between 9:1 and 10.3:1. The 89cc heads are for engines up to 124 cubic inches.

## Carburetion and EFI Delivery

On carbureted models, I use the S&S Super E with the .570” lifts, and go to the Super G for .585” lift cams and larger. Many mechanics prefer the G on the .570” lift, but I think the venturi is a little large at this stage.

I prefer the 45 Mikuni Smoothbore as it is newer technology that meters fuel more precisely in engines up to medium-sized strokers, until the esteemed S&S Super G takes over once more just because of its sheer size. Mikuni now has a 48mm Mikuni Smoothbore that may challenge the prowess of the Super G.

However, Electronic Fuel Injection is the way of the future. We cannot make the Twin Cam engine exceptionally fast within its capabilities without addressing this ideal fuel delivery.

The days of the carburetor will be pretty much over with the new emissions guidelines in 2010 and beyond as they are dinosaur technology compared with the power EFI is capable of delivering. Of course, H-D eliminates carburetor use in 2007 but some aftermarket engines meet 2010 approval with

carburetion. What isn't said is that specific restrictive exhaust and ignitions must accompany these carbureted engines to meet EPA approval.

EFI has so many advantages over carburetion. The precision of fuel delivery on EFI system lowers harmful emissions and improves fuel economy. For the sophisticated, it can also do wonders for high-performance goals.

Carburetors are finicky and imprecise. They cannot deliver the right amount of fuel and air for all rpm ranges and load demands on the engine. EFI can deliver precise amounts of gas and air at any rpm and at any load. Check out a V-Rod if you want to see what EFI is capable of in the speed department while meeting emissions dictates.

A carburetor draws and mixes fuel and air via vacuum,

manifold pressure differentials and venturi air speed. Electronic Fuel Injection delivers fuel under pressure to the injectors. Injectors squirt fuel on demand as a pressurized spray.

I address carburetion and EFI at length in Chapter IX: Fuel Delivery, sections, Tuning the S&S Super E and G Carburetor, and Tuning the Mikuni Smoothbore and EFI: Ideal Fuel and Performance EFI Ignition and Fuel Modifiers.

## EFI Performance Components

If looking at the history of injector, fuel pressure regulators and throttle body size on Harley EFI systems in use since 1995, there are interesting patterns not the least of which, some reasons why the Delphi system is so superior to the original inefficient Magneti-Marelli system.

Injectors

Magneti-Marelli injectors (27218-95), rate at 21.6 lbs per square inch (psi). In 1997, the injector (27135-97), rating increases to 30.64 psi.

The early Delphi injectors in use on 2001 models (31511-01) do increase in size once again to a 32-psi rating. Injectors on 2002-2003 FLHT's and Softails (27609-01), increase to 32.74 psi, and on it goes.

Over the years, injector size increases over 50% from the first EFI used on Harley-Davidson. However, bigger cubic inch engines require much larger injectors to feed the necessary extra fuel with the increased airflow. These are readily available now.

## Fuel Pressure Regulators

Stock fuel pressure regulators remain mostly constant at 44.3 psi over the years although aftermarkets ones are available at the time of writing (2008), in 48.2 psi, 50.5 psi and 58.6 psi.

## EFI Throttle Bodies

The size of the EFI throttle bodies also needs addressing for the bigger size engines. This component is very similar to that of the carburetor except without the air and fuel passages. The components I will list in the next few paragraphs go from what was available in 2005 to what is available presently in 2009. New stuff comes on line all the time. I do this to demonstrate how fast this area of performance is developing.

What was big a few short years ago may now be for stock use. In 2005, Horsepower Inc. (www.horsepowerinc.net) had throttle bodies for the Delphi EFI sized at 51 mm, a full 6mm bigger than the 45mm throttle body in 2005 and 53mm bodies that are 8mm larger. They accept all stock sensors; throttle cables and components including the air cleaner. Their intake manifolds have been line-bored to accept either size seamlessly for superior flow. Screamin' Eagle products now equal and in some cases eclipse this stuff that was leading edge only a few years ago.

If doing all this work, it is beyond me why anyone would use the stock restrictive cleaner.

In 2008, big bore throttle bodies available for the 1995-2001 Magneti-Marelli EFI from Horsepower Inc. are available in huge 48mm and 51mm sizes with intake runners for 1.660", 1.710", and 1.800" dimensions. They also have 62-

psi fuel regulators. The reader may not think these huge until realizing that the stock Magneti-Marelli throttle body was a very anemic 38mm.

The 2001-2005 throttle bodies are now available in 2008 from Horsepower Inc. in 48mm, 51mm, 55mm, and 58mm sizes that have intake runner diameters for 1.660”, 1.710”, and 1.800” dimensions. The 2006 to present Delphi throttle bodies excepting fly-by-wire are available in 48mm, 51mm, 55mm, and 58mm sizes that have intake runner diameters for 1.660”, 1.710” and 1.800” dimensions.

They also have their Max Flow series throttle bodies for 2001 to present Delphi models for bigger cubic inch engines that emphasize midrange to high rpm horsepower and torque. The Max Flow 1 is currently available in a 55mm throttle body to a 58mm ported intake with 1.710” and 1.800” intake runner diameters. The Max Flow II has a 62mm throttle body and intake. This is huge.

Rivera Engineering (www.riverengineering.com) has 52mm to 62mm EFI throttle bodies for big inch applications.

The Screamin' Eagle (www.harley-davidson.com), pro 50mm EFI throttle body (27623-05) with injector part number 27654-06, is for the Delphi system and is 4mm greater in diameter than the 46mm 2006 Dyna and 2007 Twin Cam (27708-06), EFI body. It fits with larger injectors (27709-06A), to match the airflow rates of the throttle body. Screamin' Eagle also currently has a 58-millimeter Delphi throttle body with matching injectors.

Now let us look at what S.E. has available for the smaller 1995-2001 early Magneti-Marelli EFI. It is a mere 43mm diameter throttle body compared to the stock 46mm body used on the new Twin Cams but it is larger than the stock

Magneti-Marelli (MM), 38mm body.

All Delphi EFI injectors squirt their pulse-widths at the back of the intake valves, as does the S.E. 43mm upgrade for MM.

Injector sizing, throttle body diameters and fuel pressure increments are changing all the time. A well-versed mechanic with expertise in these areas becomes invaluable.

JIMS USA (www.jimsusa.com) has an elliptical venturi throttle body that promises a quick throttle response with maximum torque and horsepower throughout the rpm range. The Ellipse 58mm throttle body has a wider manifold for taller engines. This throttle body is available with or without electrical components, the MAP sensor, fuel rails, and injectors. The design is for 100 cubic inches to 131 cubic inch

engines.

JIMS, of course also supplies a 131 cubic inch engine.

S&S Cycle, Inc. has a comprehensive selection of EFI components for big inch strokers. There are 5-different sized throttle bodies and injector systems for performance Twin Cams from the stock displacement 88-incher to 95-inch, 100, 106, 107, 116 and the massive 124-inch displacement. The high flow injectors come with the S&S manifolds that match to the throttle body chosen.

S&S also provides their variable VFI (Variable Fuel Injection) modules for either MM or Delphi. The stock fuel pump, regulator and fuel lines are adequate for usage up to and including the 124-inch engine. It is possible to get around 135 HP out of their 124 kit. Again it takes a skilled mechanic to finesse this, as coordinated support parts are where it is at.

The VFI system, which replaces the stock ECM, automatically adjusts for altitude variations. It uses the five stock sensors; crank position, MAP (manifold absolute pressure), throttle position, air temperature, plus cylinder head temperature. It also uses 02 oxygen sensors that self adjusts fuel delivery in real-time via analysis of the exhaust gases. S&S sells two oxygen sensors, one for each pipe, with weld in exhaust bosses and a wiring sub-harness.

Their high flow manifolds in different styles and sizes accommodate different length cylinders. They have independent runners that tune to the throttle body for maximum airflow.

Lastly, there is an S&S tuned induction assembly with a separate tuned intake tract for each cylinder. Their design

follows physics principles in terms of length, diameters, and tapers that utilize air column inertia, and timed kinetic pulse gas dynamic pressure waves. A natural supercharging effect takes place whereby additional air rams into each cylinder for additional power. The forward swept design of the intake runners improves power via the ramming of cooler and therefore denser air than is taken in via a traditional air breather.

Once again, this area of performance is changing monthly with new products from top quality companies. Next, I am going to look at a performance impediment and the cures to ensure big-inch longevity and reliability.

## Performance Sprocket Shaft Bearing Problem

The left side engine-case-bearing(s) support the left flywheel sprocket (power), shaft.

Early Twin Cams from 1999 to 2002 use OEM 9028 bearing sets comprising of inward-facing tapered Timken bearings consisting of two races and two bearings with a crank end-spacing shim between them. The shim determines endplay of the flywheel assembly. Endplay is how much the flywheel assembly can float back and forth inside the engine casings. The sixteen different sized shims vary in size from .0895” to .1195” in successive gradients of .020 thousandths-of-an-inches. Successfully used in all Big Twins from 1956 to 2002 these long-lasting bearings kits are very durable.

The OEM is the acronym for Original Equipment Manufacturer…parts supplied by the manufacturer of the vehicle, in this case Harley-Davidson.

OEM 24604-00, B, C, and D bearings are a big roller

bearing measuring 2 OD with an ID of 1 In my opinion, these roller bearings are well suited for the up and down vertical pressures imposed both on the pinion and sprocket side shafts of the crank extending through the engine casings.

However, I feel they are ill suited for the lateral horizontal-pulsing power surges traveling sideways from the crank to the primary compensating engine sprocket.

## Twin Cam Sprocket Shaft Bearing(s)

The 1999-2002 engine casings that utilize the opposed and tapered Timken bearing sets to support the sprocket shaft in the left casing successfully since 1956 are a superlative design. In my opinion, these 360 special formulated proprietary alloy casings, with care will be serviceable for a

half-million miles and beyond.

The 2003 and later casings will not be as fortunate since the engineers have eliminated the very reliable sprocket shaft Timken bearing sets in favor of a single roller bearing. The year 2000 and later Softail engine right side pinion-shaft roller bearing has supplanted the left side Timken bearings on all 2003 and later Twin Cam engines. I do not feel this type of bearing that works wonderfully with the up-down stresses imposed in the right casing will be able to withstand the extra rigors forced by the throbbing sideway power pulses of the transferring power stroke down the powertrain to where the rubber contacts the asphalt.

The Timken bearing sets are more tolerant of crankshaft flexing at high rpms.

Interestingly, both current manufacturers of aftermarket Alpha Twin Cam casings use the Timken bearing set up instead of the roller bearing on the sprocket side for all year applications. JIMS USA (www.jimsusa.com), also makes Beta Softail Twin Cam cases that use the sprocket shaft Timken bearing sets for all years also.

However, if using 2003 and later stock engine cases for a performance application that use a roller bearing on the sprocket shaft do not let this bother you, as I will describe two ways to overcome this potentially bad performancing situation in the next section, 2003-Present Twin Cam Engine Casing Solution.

I have heard reports that some year Evo casing halves were machined separate to each other in different operations

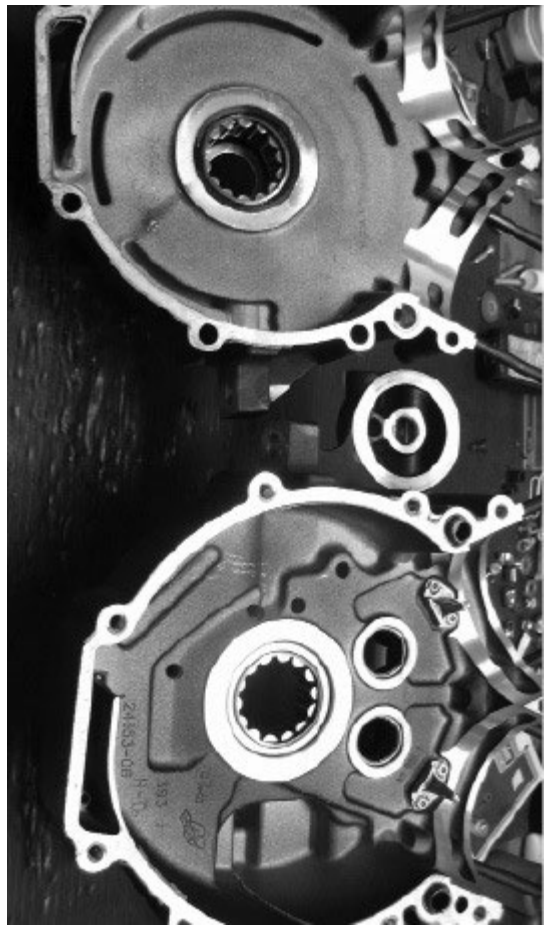
(locations), so that sometimes the mating gasketing areas did not match planes with each other with resultant leakage. This is not true for the superlative Twin Cam casings. I have already discussed that the Twin Cam casings are only clamped five times reducing machining distortions from the previous 32-clampings necessary for producing Evolution casings.

As mentioned, earlier and as testament to their quality, only two companies, S&S Cycle, Inc. being the first and now JIMS USA in the aftermarket make replacement casings for the Twin Cam. The public may purchase S&S Alpha casings. Their purchase is available through any S&S Cycle, Inc. dealer like my shop, Heavy Duty Cycles Toronto (www.heavydutycycles.com). A dealer locator is on the S&S website.

Both companies offer excellent quality.

However, the purchase of JIMS Twin Cam casings and big inch Twin Cam kits must be through an authorized Harley-Davidson dealer. It gets better as only licensed AHDRA racers are able to purchase the casings by themselves

Bearings Inside Engine Casings



Pictured are the inside engine casings for a 2007 CVO 110. The 2003 to 2006 TC88 and the 2007 to present TC96 cases are much the same except the cylinder spigot holes are smaller. The upper case is the left-hand sprocket shaft side. The sprocket shaft extends perpendicular from its power side flywheel and protrudes through the centrally located 24604-00D roller support bearing in the engine case into the sealed primary system that connects the engine to the transmission. The sprocket shaft carries the alternator rotor whose internal shell magnets rotate around the fixed stator bolted to the outside of the motor casing housed inside the primary. The compensator sprocket package fits on the sprocket shaft next. Then the primary chain fits onto this sprocket transferring power back to the clutch sprocket inside the rear of the primary. The compensator, a series of heavy diaphragm springs, absorbs shock pulses to protect the rest of the drivetrain. What is dramatically different in these engine

casings from 1999-2002 TC88 ones is the use of a roller bearing on the sprocket shaft side instead of the traditional OEM 9028 opposed-tapered Timken bearing sets. The Timken bearings have served H-D engines reliably and efficiently in one form or another since 1956 through the Panhead, Shovelhead, Evo and the first years of the Twin Cam eras. The 9028 dual tapered bearings and races face each other, separated by a spacer that sets crankshaft endplay. These Timken bearing sets ably withstood the power pulses throbbing out the sprocket power shaft for transfer back through the powertrain to the rear wheel. The bottom picture is the right side pinion-shaft engine case. The pinion shaft extends perpendicular from its right side flywheel and protrudes through the centrally located 24604-00D-roller support bearing in the right-side engine case into the timing chest. Inside the timing chest, the protruding pinion shaft supports and rotates the geroters inside the oil pump housing that bolts to the cam support plate. The support plate distributes oil fed in via the 5-micron oil filter throughout the

engine. Oil pressure feeds from the oil pump into the support plate to also feed the bottom end crank bearings, crank pin and con rod races, via its pinion bushing into the internal pinion shaft oiling passage. The last part of the pinion shaft extends through the support plate pinion bushing to drive the rear cam, front cam and their respective valve trains via a slotted drive sprocket fitting on a mating pinion shaft slot and bolted firmly to its end. The two roller bearings one on either side of the flywheel assembly support the rotating flywheels and their reciprocating connecting rod components and pistons. The right shaft supports and powers the oil pump as well as driving the valve train(s). The left shaft transfers the power generated by the descending piston's power stroke back into the powertrain to the rear wheel. The question becomes whether these same two huge 24604-00D roller bearing up-and-down vertical pressures imposed both on the pinion and sprocket side shafts of the crank extending through the engine casings. Therefore, this bearing is suitable for the pinion side and serves it well. However, I feel this type of roller bearing is

ill suited for the lateral horizontal-pulsing power surges traveling sideways from the crank to the primary compensating engine sprocket. The 9028 tapered Timken bearings are best for these lateral types of stress. Above the lower case pinion roller bearing are the backsides of the inboard rear-cam-journal-bearings. The rear cam journal INA SCE168 (OEM 9195), support bearings in Harley-Davidson's judgment is not a problem. I differ with the Factory perception as we have just taken too many Twin Cams apart for performance reasons, only to find what I consider adverse wear on these types of bearings. Furthermore, many of these engines had minimal mileage. S&S Cycle, Inc. and Crane Cams supply replacement Torrington bearings with their cam-gear-drive kits to replace stock INA bearings. At Heavy Duty Cycles, we replace the stock INA SCE168 (OEM9125), bearings with Torrington B168s in our entire Twin Cam engine work in the 2006 Dyna and all 2007 and later models. H-D stays loyal to the 1999-2006 INA SCE148 (OEM 9198), style bearing that we replace with Torrington B148's. H-D

increases its size from the 1999-2006 .875” inner diameters a full .125” to a new and bigger 1.000” inner diameter. This bearing is definitely stronger than what went before by virtue of its size is perhaps a design admission by the Factory that they, too, must have seen adverse wear on the earlier smaller INA bearings. Looking to the right and left of the cam journal INA bearings near the mouths of the cylinder spigot holes are a full view of the front and rear piston oil cooling jets. Above and connecting to the lower casing in the picture is the oil filter housing. A 5-micron (5-millionsths of a meter), filter attaches to this housing, cleansing the oil before routing it back into the crankcase and the timing chest cam-support-plate for distribution to the crankshaft through a feed bushing in the support plate. Oil fills a main galley in the cam support plate to supply oil to the lifters and the top end. Two small oiler holes squirt oil onto the two timing drive chains. Oil then travels to the piston-cooling jets, which squirt a stream of oil (35 psi at 230° F at 2000 rpm), from their location at the base of the engine cylinder deck. This cooling action

relieves about 50° F from piston head temperature. The sump plate to the left on both casings joins in assembly. Oil drains into this area and sumps back into the oil pump in the timing chest for return to the oil reservoir. This allows for more efficient oil return and drains away oil from the churning flywheels. If oil picks up by the flywheels inertial drag results.

# 2003-Present Twin Cam Engine Casing Solution

There are currently two ways of converting 2003 to present left-side Twin Cam engine cases to use the 9028 Timken tapered bearings. For big-inch engines, it is a very good idea to convert away from the single sprocket shaft roller bearing whether it be 24604-00B, 24604-00C, 24604-00D or the last current one 24605-07 to the 9028 Timken tapered kit.

Zippers Performance (www.zippersperformance.com) was first off the line. They offer a machine shop conversion service for left-side casings. It is necessary to provide the casing for the conversion.

STD Development Company (www.stddevelopment.com),

the very respected engine case and performance head manufacturer will also take the left ball bearing engine casing, machine it to accept the old style and very reliable steel insert along with supplying the Timken bearing package.

JIMS USA makes a conversion tool for a 9028 bearing kit installation into 2003 to present Twin Cam left-hand casings. This tool (part number 959) eliminates the need to send cases out for Timken bearing machining. The JIMS 959 tool, installs a sleeve, which will accommodate the Timken bearings directly into the cases using a 2-ton press and simple hand tools. The steel insert is several times stronger than the aluminum case material the standard roller bearing rides in.

There is one potential problem that the astute mechanic performing the bearing conversion will check and that is to ensure the engine casing bearing holes are in alignment with

each other. Duhh, H-D does this when manufacturing the engine cases! Well, they certainly used to. I think that using the roller bearings on both the pinion and sprocket shafts allow extra play if the bearing holes do not align perfectly. The Timken bearings do not allow misalignment. If the bearing holes do not align and a Timken conversion is used then longevity problems will certainly ensue. One problem will be correcting while creating another equally dangerous situation. Therefore both engine casings need to be indicated, which is machine shop vernacular for measuring for alignment. If out of alignment then a machine shop process called line boring is in order before installing the Timken conversion.

Of course, using 1999 to 2002 H-D Twin Cam casings or S&S aftermarket ones will resolve the problem from the get-go.

The increase in swept volume with stroking, boring, or both creates displacement of more air as the pistons ascend and descend in their respective cylinder bores. The churning internal engine components, particularly the rotating flywheels, create an oilified gas misting that mixes with the displacing air. Displacing air with its oilified component pushes through the pinion bearing into the timing chest and then up the pushrod tunnels into the rocker wafers by the descending pistons. The air must have access to outside venting. Here, the breather apparatus must separate the heavier oil from the air, allow the oil to drain back down into the timing chest via gravity while allowing the lighter air to vent to the outside world through a vent in each head into the air cleaner.

This subject is dealt with in detail in Volume I: The Twin

Chapter IV: Harley Oil Leaks: Marking its Spot in many subsections beginning at Puking Gravity Feed Oil Tanks and ending with The 2007 Harley Scavenging There are also numerous pictures of the various oil separation techniques in Volume I.

I will review the various methodologies and any updating since writing the last volume.

Blowby, and oil carryover are terms used to describe different phenomena of oil escaping the engine other than through a gasketing leak. The basic idea or concept in each has commonality. Quite often, the terms are interchangeable and are synonymous with each other. Sometimes they refer to different phenomena. Read the terminology within the context of use.

Blowby

Blowby or carryover means that a gaseous mixture, liquid gas, or oil penetrates an area from which it is restricted. One example of blowby is with oil overwhelming piston rings and blowing past them into the combustion chamber. Oil has no business in the combustion chamber.

## Venting

Venting usually has to do with the physical principle of a vacuum; for example, where air needs to replace a fluid draining out of a sealed container. Gasoline draining out of a gas tank to feed the fuel system's carburetor or pumping to injectors or oil, leaving an oil tank to lubricate the engine has much the same vacuum principle to deal with. A vacuum creates as the fluid leaves its holding tank since a cap seals the tank. Air must fill this vacuum or gas/oil flow will impede. In

this case, the tank will vent by air filling the vacuum left behind by the departing liquid.

Engine venting is necessary where internally moving parts displaces air. Some air vents out of the engine. Other air vents back in to fill vacuous areas vacated by parts moving in other directions. The best example of this is ascending and descending pistons displacing air in the bottom-end compartment, into the timing chest compartment and up into the top end engine compartment. Blowby may also mean the transfer of oil out of the engine into the outside world. The gases comprise mainly of air venting out of the engine.

The Breather Apparatus

Sometimes there is oil vapor in this air that liquefies and deposits into the air breather to drain out of it onto the engine's side. This is a form of blowby resulting from vented

gases.

Any increase in engine displacement or sustained high rpm riding or both may overwhelm the breather apparatus located in each rocker box at the top of the engine with liquifying oil mist into oil. In these cases, the oil carryover or blowby into the breather will spill out onto the outside engine and in extreme cases, the rider's right pant leg.

Modern Engine Venting Breathers

In 1992, the Factory mercifully relocated the vent to two positions; one in each head, where the umbrella valve and separation chambers allow displaced air-vapor to escape into the top of the air cleaner. These vapors then suck into the combustion chamber for more complete combustion in the best attempt up to that time in controlling harmful emissions

and annoying oil deposits. The only problem is when there is inadequate air and oil separation, Harley once again makes its mark, all over the side of the bike.

The Twin Cam 88 and the TC96 attempts to address this ongoing problem with a three-stage oil/air separation unit.

The TC96, because of previous blowby experience in its predecessor TC88 and an increase in displacement also receives a newer style oil pump that scavenges 23% more oil out of the engine for return to the oil tank in an attempt to further control oil carryover.

Each breather-separator locates inside the wafers on top of the heads between the rocker arms. After separation, the air vents along a convoluted path into the air breather. Preceding separation, the oily mist travels through the one-way umbrella

valve into a depressed separation area where gravity once again assists in further separation of the oil and the air. Theoretically, the air goes out through a raised hole down into the carburetor breather. This vent is seen externally as the two big bolts that screw into the heads, right and left, near the top of the inside breather plate and into the air box. The bolts are hollow to allow the gases' flow-through access.

## Repair the Cause not a Symptom

Using the wrong sealing O-rings, improper installation, or neglecting to check for misaligned or warped parts inside the engine often causes blowby. Using an exotic solution for this situation is attempting to correct a symptom instead of the cause. It may partially fix or even totally fix the symptom, but why layer the situation? The first areas we check at Heavy Duty Cycles when we encounter blowby is that all the stock components that have an ability to cause this situation have correct fitment, are not damaged, or missing.

Yes, we have found missing sealing O-rings not installed at all, both apparently right from the Factory assembly line and by neglectful mechanics in dealer and aftermarket service shops! It is of paramount importance that the O-rings that act as seals between the many component parts of the oiling system are not leaking. If various O-rings are not able to do their job, particularly on the oil-feed side as intended by Harley, then oil will seep, bleed, or gush past the juncture and into the timing chest compartment. This is tantamount to internal bleeding, not only causing a lowering of oil pressure but a situation the venting system was not designed for and therefore not prepared to deal with.

The workshop manual also addresses these issues and cautions against distorted, wrong, pinched, or missing O-rings. It also warns that some different O-rings may indeed look the

same and used inadvertently. Oil leaking past a worn or forgotten seal can overwhelm the return side of the oil pump, as too much oil will build up in the cam chest compartment. The return side of the pump cannot remove this excess oil fast enough because it is not designed for this much quantity although the TC96 increases oil scavenging by the aforementioned whopping 23-percent.

Some of the buildup of oil starts to vaporize and mixes with displacing engine air. People, and that certainly include mechanics; generally forget to check the obvious. A mentality exists that there must be a more complicated cause to the perplexing problem at hand. The following are some necessary pre-checks to isolate the cause of a venting problem.

- The most obvious and easy thing to check in this case is to ensure that the oil pan/oil tank is not overfilled. The tank must have air at the top to allow venting. No air equals no

venting. No venting means that excess oil will force out somewhere.

- Secondly, check a suspect venting problem for a clogged oil return.

- Next, inspect all oil-feed junctions that seal with O-rings.

- The O-rings used extensively inside the TC88 engines must be the correct ones.

Some look like others but are not quite the same. Using O-rings that seem to fit okay but are not exact can cause problems like blowby.

- Then, ensure the correct installation of the O-rings.

- Finally, guarantee the O-rings are capable of sealing. Do this by inspecting mating parts for defects or warpage, thereby preventing the O-rings from effectively sealing.

However, who is to say, as I have serviced a bike with a crucial O-ring missing that did not suffer blowby. I can only surmise that the machining on both mating parts must have been superb, the bike had not been bored to 1550cc, and the rider rode very conservatively.

In one blowby-plagued machine that we serviced, we discovered that the oil pump surface that mates against the cam support plate was mal-machined right from the get-go at the Factory; the surface was irregular as if there were a machining problem. Replacement of the support plate solved the blowby problem.

Oil Feed O-Ring Junctures

Examination of all junctures is necessary and O-ring installation performed correctly.

Some crucial areas are;

- The side of the oil pump that fits against the cam support plate is the feed side. This O-ring 11286 seals the pump against the support plate.

- O-ring 11301 seals the cam support plate to the engine casing and on TC88B engines.

- There is an extra feed for the counterbalancers. It also takes an 11301 O-ring.

If this O-ring is missing, crimped etc., oil will bleed down into the timing chest.

The cam support plate from 2000 on has this feedhole.

- When installing the cam support plate on a TC88Alpha model without counterbalancers, there is a screw and washer to block this hole. If not blocked, oil will feed right into the timing chest unobstructed with resultant oil blowby, plus a loss in oil pressure.

An oil pressure drop at highway speeds will be noticeable with any of the above seal leakage examples because of a lack of oil pressure. Lifter noise is a sure sign that this may be happening. Always check oil pressure at highway speeds to determine that leakage may be occurring, which will cause oil build-up increasing the potential for blowby.

Blowby does not happen immediately because the build-up and resultant actions take some time. This is why blowby sometimes does not occur until after sustained high-speed riding.

Oil Pump Alignment

Oil pump alignment is frequently out because of improper installation. This can result in premature wear as well as an improper seal against the cam support plate.

- On the cam support feed side, this allows oil seepage into the timing chest and possible resultant oil carryover.

- On the crankcase return side, an improper seal might

result in less than ideal oil return from the crankcase flywheel compartment.

Align the oil pump by,

1. Firstly, lube the O-rings that fit between the pump and mating parts, the crankcase and the cam support plate, with a thin film of clean 20W50 engine oil.

2. Assemble the oil pump loosely against the backside (engine side), cam support plate, and slide over the pinion shaft.

3. Install the cam support plate to the crankcase flange according to the manual with the six 1”-long screws.

4. Torque the screws to 90-120 in-lbs, metric: 10.2-13.6 Nm on the TC88 pump. Do not make the mistake of doing this

in ft-lbs, since all the screws will strip the threads in the softer aluminum-crankcase flange.

5. The bolts torque down in the following sequence: bottom left; right-side upper; top left; right-side lower, left side, and top right.

6. Alternately, snug and tighten a few times until reaching the specified torque.

At Heavy Duty, until we correctly align and torque the oil pump to its specified values, we do not tighten the cam support plate to full torque values. Pay attention to Allen head socket screw length since both the hydraulic lifter covers and the cam support plate utilize the same ¼”-20 threads per inch screws. The lifter cover also uses Allen head socket screws that are ¼”-20. However, the lifter cover uses screws (OEM

4741A) that are 1 ¼”, while the cam support plate uses 1.000”-screws (OEM 4740A).

In 1999, there were no specialized H-D tools for oil pump alignment even though Evo mechanics had one right under their nose.

1. Mechanics would install the four oil pump bolts loosely to keep the oil pump semi-aligned.

2. Then slowly rotate the engine to align the pump. The oil pump seeks the most comfortable alignment spot.

3. Gently snug the bolts.

4. Then rotate the engine again. A sensitive mechanic will be able to feel inappropriate drag if the oil pump is not aligning correctly.

Now torque the oil pump bolts to indicated values (90-120 in-lbs; 10.2-13.6 Nm), and in the proper sequence. Cast into the cam support plate are the four sequential numbers; one is bottom left; two is top right; three is bottom right, and four is top left.

5. This alternative snug-and-tighten methodology is utilized everywhere in mechanics where there are a number of fasteners holding down warp sensitive parts.

Then someone made a discovery that the Evo 1984-1999 Big Twin hydraulic lifter guide, also known as a tappet guide block alignment tool (HD33443), would also align the Twin Cam oil pump if an extra tool were purchased. There is no excuse, unless on the side of the road, for not using these two alignment tools. They are 1 ⅝" long, with one end a ¼"-20-

threaded portion to screw in place of the oil pump hold down screw while the other end has a hex head to accommodate a five-sixteenths inch wrench or a five-sixteenths hex head socket. The middle portion has a graduated smooth taper that increases from the quarter inch to five-sixteenths inch.

1. Using two of these tools diametrically opposed to each other on either side of the oil pump centers the oil pump to the holes in the cam support plate perfectly, thereby aligning the oil pump.

2. Leave the two oil pump-screws loosely in the number three (bottom-right), and number four (top-left), positions.

3. Remove number one (bottom left), and number two (top right), screws and insert the two alignment studs in their place.

4. While slowly rotating the engine alternately snug the alignment tools until tightened to 40-45 in-lbs, which is a little less than half the final specified torque.

5. Then alternately snug the bolts in hole number three (bottom right), and number four (top left), positions until tightened to the same intermediate 40-45 in-lbs.

6. Install the bolts, alternately snug, and tighten, again to 40-45 in-lbs after removing the alignment pins.

7. Finally, tighten all for screws (bolts), alternately in the sequence described above to the final specified torque values.

However, even when correctly doing all of the above there are still Twin Cam engines that will expel oil with sustained high rpm riding and/or bigger cubic inches. Any of the exotic

aftermarket fixes that we discuss in these articles are well worth doing and a definite upgrade to the engine in terms of breathing functionality. First, let us look at a simple modification that will assist top end oil drainage in any year Twin Cam.

Enlarging the Drain Hole to Assist Venting

A necessary trick will help if blowby is problematic or if the Twin Cam heads are off the bike for any reason.

The separating oil in the middle-wafer-depression-area will flow via gravity through a small drain hole.

- While double-checking the umbrella valves, H-D recommends drilling out the drain hole to drill number size-42, which is ninety-three thousandths-of-an-inch (.093”)

in size.

- I recommend drilling this hole out another ten-thousandths of an inch to .103”.

- I sometimes go up to one hundred and ten-thousandths of an inch (.110”), with this sizing on problem bikes. I do this to .110” on my own bikes with great results.

I do not advise going any larger as it is easy to make a hole larger but very difficult to decrease its size.

This simple assist helps ensure there is no build-up of oil waiting to drain that might overflow into the breather hole and down into the air box and out all over the engine.

The Real Problem

In my opinion, H-D has not addressed the real problem, which is air, displacing by the descending pistons disrupting return oil flow from the heads back down into the crankcase. Displacing air also travels up the oil return holes from the heads, sometimes disrupting the returning oil causing a backup. This causes an oil build-up in the heads that, not always but sometimes, overwhelms the breather apparatus that is supposed to separate the oil and air for venting purposes.

Now, it is easy to understand why increasing cubic inches can be an aggravating factor since more air displaces because the piston cylinder bore is larger in diameter and/or the pistons are traveling further up-and-down the cylinders. Sustained high rpm creates the same type of problem since there is a more rapid piston movement of displacing air to

disrupt more return oil flow, causing a potential backup in the heads.

## The S&S “Reed Valve” Solution

The S&S Cycle, Inc. (www.sscycle.com), solution is a crankcase breather valve for the TC88A or Alpha motor. There is not a reed valve available for a TC88B or Beta engine or Sportsters because of space considerations. The TC88B has no space because of the counterbalancer assembly.

The reed-valve-breather-apparatus is a machined billet aluminum body containing four steel reed valves. The reed valve installs easily, especially if already performing cam work or checking the oil pump.

The one-way-flow reed valve allows the escape of air from the crankcase into the timing chest but does not allow it back

in. The displacement cycle is broken, allowing oil to return to the timing chest. The reed valve design works on crankcase pressure changes or differentials. This upgrade complements the stock breather apparatus located inside each rocker box.

The four steel reeds are normally in a closed position. When the pistons descend in their cylinder bores, the air in the crankcase below the pistons becomes increasingly compressed. The descending piston displacement pressure flows between the ball and bearing seats in the massive pinion roller bearing that supports the right-flywheel-pinion-shaft into the timing compartment. The Twin Cam's approximately three-inch circular donut-shape reed valve that is about three-quarters of an inch thick, sits on the pinion shaft behind the internal oil pump and in front of the pinion bearing separating the two compartments; the timing compartment from the bottom-end compartment.

This pressure build-up of displaced air from the descending pistons forces the four reeds to push open, thereby allowing the air and oil mist in the crankcase to flow and escape into the cam timing chest compartment.

When the pistons change direction and ascend their bores, a negative pressure creates in the crankcase. This vacuum pulls the four reeds into their closed position, preventing air from entering the crankcase compartment via the pinion bearing. Oil scavenging greatly improves by the effect of this slight vacuum, which reduces the potential for blowby.

Since the reed valves open and close precisely on demand, they automatically compensate for the mechanical lag time associated with the accelerating pressurized air mass as it forcefully exits out of the crankcase. This is most significant at

higher rpm where blowby potential is at its greatest when the time available to get the air out is the shortest. It also compensates for faster moving pistons in much the same way in big-inch strokers. Therefore, air displaces faster increasing the potential once again for blowby. Big-bore kits also result in displacement of a bigger mass of air.

The reed valve is a huge help in assisting the breather apparatus in separating oil from air and allowing the air to escape into the breather, only to draw back into the engine's combustion chamber to re-burn. The bottom line is that the moving gases caused by the pistons descending and ascending in their respective bores with the resultant pressure differentials open and close the four reed valves in tandem, allowing for accurate and precise timing no matter what the changing piston speeds are.

In fact as S&S says in its 2007 Performance "Our engineers were able to determine that the problem was caused by the

continuous rise and fall of air pressure in the crankcase, cam chest, and rocker boxes. Large volumes of air moving in and out of the crankcase cause pressure changes as the pistons travel up and down. Because of this, airflow and the changes in pressure in the cam chest, air also flows in and out of the rocker boxes through the cylinder head oil drain passages. At sustained high speeds, the constant in-and-out flow of air in the rocker boxes interrupts the free flow of oil from the cylinder heads to the cam chest.

“In fact, a small quantity of oil, pushed up and down by the pressure changes, is trapped in the drain passage, effectively blocking the passage so that no oil can drain out of the rocker boxes.

“When enough oil has accumulated in the rocker boxes, it will start to blow out the breather passage to the air cleaner and ends up all over your bike, your passenger’s new boots, and, in extreme cases, onto your rear tire. The reed valve allows air and oil mist to escape from the crankcase, but do not allow them to re-enter.” 1 S&S goes on to relate, “An engine with the S&S Breather Valve still has large swings in crankcase pressure as the pistons travel up and down, but that average crankcase pressure is reduced….The elimination of pressure changes and resulting airflow in the rocker boxes, allows the oil to drain smoothly down to the cam chest without interference from the turbulence. No more oil carryover.” 2

S&S Proprietary Rocker Boxes and Gasketing

The S&S rocker boxes use an exclusive one-way valve built into their rocker cover gasket to control oil carryover in head

breathing engines.

## The TP Pro Vent Solution

If upgrading the breather apparatus in order to better separate oil from venting gaseous air and/or to handle bigger volumes pushed up into the rocker boxes, the possibilities of controlling blowby improve. The development of Pro Vent rocker boxes by Tom Pirone of TP Engineering (www.tpengineering.com) has built-in breather components with a view to curing blowby on late Evo, mid-year XL's, and, both Alpha and Beta Twin Cam engines. I think it obvious that one of the reasons that companies like TP and S&S are at the forefront of developing ways of controlling oil carryover is that a main part of their business is the manufacturing of big-inch V-Twin engines.

The TP ProVent two-piece rocker boxes bolt onto the above model years without any modifications. The TP Engineering two-piece rocker box is innovative in design and function. The gorgeous CNC-machined from billet aluminum mirror-finished boxes have a sleek appearance and complement any top end. Extra wall thickness reduces valve train noise.

TP Performance reengineers the way oil separates from the venting air flowing out from the engine into the breather. Easier air movement separates the air from the oil more efficiently and nearly eliminates oil carryover into the air filter and all this without the use of rubber flapper valves found in Evo and Twin Cam rocker boxes.

To accomplish this,

- TP utilizes a series of machined labyrinths to collect oil

droplets as breather gasses traverse the passages at high speeds. The easiest way to solve mechanical problems is to use the physical properties of matter; solid metals, liquids, or gasses to achieve goals, as opposed to having their physics work against you. Harness nature instead of fighting it.

- Displaced oilified air enters into the first chamber of the Pro Vent system via three entry holes that begin the process of extracting oil droplets from the vapor.

Each surface touched by the breather gasses collects suspended oil. The walls are relatively cooler lessening the air's capability of carrying suspended oil.

- The air collides with the walls of the chamber and makes a series of turns before climbing the last wall and flowing through the entry hole into the next chamber.

- Rapid directional turns, again lessens the air's ability to suspend heavier substances like oil because of inertia, momentum, and gravity.

- In the second chamber, the breather gasses spin and collide with the walls, and by the time the air climbs the final wall on its way out of the chamber, it is free of oil and ready to enter the crossover channel in the rocker box cover, which routes the air out of the head and into the air cleaner.

In fact, TP claims this system cleans the air leaving the engine so well it allows for the elimination of the timed rotating breather valve in Evo engines!

Crankcase pressure reduces by 2 psi.

At Heavy Duty Cycles (www.heavydutycycles.com), we

find that the TP Engineering Pro Vent rocker wafers worked when we tested them on problematic engines. They are trick and beautiful.

## Crank Vents

Crank vents can manipulate engine vent breathing. They not only prevent oil from venting to the outside world but also add a small amount of horsepower. Anytime an engine can run with less internal resistance, there will be a gain in horsepower and torque transferred back to the rear wheel although these gains are small.

## The Doherty “Myst” Free Fix

Tim Doherty at Doherty Machine (www.dohertymachine.com) has been developing various

effective systems to combat oil carryover for many years now. Tim reinvents the breather apparatus that fits in each Twin Cam head. His only concern is that the potential O-ring problems and possible oil pump misalignment are checked and addressed, if need be, before using his product. Doherty calls his breather apparatus the Mystfree.

Tim Doherty thinks, and I agree, “The two biggest reasons for excessive oil blowby are incorrect alignment of the oil pump and poor oil separation in the cylinder heads’ breather valves. Inexperienced technicians who are installing camshafts and do not follow the correct oil pump alignment procedure when they reinstall the cam plate are responsible for most of the oil pump alignment problems. The poor oil separation up in the cylinder heads is caused by the stock breather valves being overwhelmed by the volume of oily/air mist that gets up to the valves during a really hard ride.” 3

In testing the superbly made and aesthetically pleasing

Doherty Machine products at Heavy Duty Cycles, I find that they work very well.

First, remove the stock H-D breather apparatus from inside the rocker box on each head. Replace with the Doherty breather, again one for each head. Disruption of the oilified mist is essential in a variety of ways to allow physical principles like gravity to do the oil separation work.

- The first disruption of the air/oil mist occurs as it engages the copper base gasket of the Mystfree breather device.

- The oilified gaseous mix swirls around the lower cylindrical base of this aftermarket breather apparatus, which begins to separate the air and the oil.

- The stock factory units have a single chamber separator that fits with a gauze element, whereas the Mystfree breather valves use a two-chamber oil separator.

- Augmenting the first of the two chambers cast into the center of the rocker arm support is a cylinder that protrudes into it from the bottom of the Mystfree valve body.

- The first cylinder has three openings spaced around its edge that reverses the pressurized air/oil mist that is inside the motor to change direction several times, as it pushes towards the exit in the breather valve. Most of the oil separates in the first-stage lower chamber but as with the Factory H-D design, one chamber is good for problem-free Twin Cams but is not enough for problem bikes.

- As the oil mist, moves up against the walls of the lower

chamber, most of the oil from the oilified mist separates and returns via the bleed holes in the bottom of the chamber.

This procedure involves some principles of physics that we all ignored in school but need now is best explained with an analogy. The Doherty Machine Mystfree oil separation is analogous to how grass collects dew as air moves through the foliage. The grass works as a separator of dew from the air.

- When the pressurized air passes through the six holes in the Mystfree valve, it hits the bottom of the neoprene umbrella valve, causing it to rise up, thereby opening into the second chamber.

- Any remaining oil that has not separated passes from the lower portion of the unit through the umbrella valve. The Doherty umbrella valve differs from the Harley one.

The umbrella does not lock into its seat like its H-D counterpart. Instead, it floats on its stem, opening, and closing much like a piston. The floating Doherty neoprene umbrella has a smaller diameter than that of the stock Twin Cam fixed umbrella valve.

- A higher vacuum develops in the second separation chamber that helps to pull any oil back into the cylinder head's rocker box.

- This oil now drains off through incorporated into the top of the Mystfree body.

No oil residue is supposed to reach the raised air outlet located on the top front portion of the Mystfree apparatus. Two things occur with any oil that is present in the top cavity of the Mystfree.

- There is a washer with a snap ring with a .750” (¾”), inner diameter hole in it. The purpose of incorporating this is to continue the air and oil separation process of any liquid that gets past the umbrella valve.

- Creation of a vacuum under the washer sucks any oil back through the .060 vacuum portholes on the bottom of the Mystfree unit. The oil returns to the two return holes in the H-D rocker arm support on the down strokes of each piston.

This recovery oil returns to the rocker box cavity and flows downwards into the oil return hole from the head, through the cylinder and engine casing into the timing chest.

- Lastly, there is a raised stanchion in the front of the Mystfree body. If there is any liquid still present, it has to overcome that raised portion to exit the breather system.

The Power Vent

Two other nifty items that Tim sells are the Power Vent crankcase ventilation systems.

One is a cylindrically shaped single unit. I would use this one for 1991 and earlier Evolutions, Shovels and Pans, as well as earlier Sportsters that vent out of the lower portion of the engine crankcases. It will also work on top-venting engines like 1992 to present Big Twins and 1991- present XL's. If using it for this purpose, also purchase an aftermarket kit for mounting the Power Vent. Both vent holes join to a single line that connects to the single crank vent. This crank vent also comes with a nipple for insertion into the end of the vent body for attaching a supplied clear hose along with a filter. The filter is a separate purchase.

Tim Doherty has also developed gorgeous crankcase vents that fit on both the vent holes, one coming out of each head for the top venting engines. These can also attach to a filter(s) via supplied nipples. There are kits developed for 1991 and later XL's, 1992-99 Evos, and, of course, a kit for the Twin Cams. They are machined billet and polished to a lustrous shine.

Some aftermarket engines like S&S may come with both top venting heads and the lower crankcase vent. This makes it easier to control blowby because these engines are usually mega-cubed. For these applications, use the cylindrical single crank vent for the bottom and the dual kit on the top head vents.

As Tim rightly and honestly says, "One of the most

important reasons and benefits in the function of the Power Vent is that they significantly reduce oil carryover. If installed on an engine with minimal oil mist they have proven to reduce the mist to carryover to almost nothing. This occurs as of a result in the decrease in crankcase pressure. However, I would like to make one point clear. Other manufacturers of similar products have advertised and claim these types of externally installed devices will stop oil misting. No external device will accomplish this “period.” 4

With a problem bike, the Mystfree works much better to solve blowby or carryover.

The Power Vent is a very air-sensitive, one-way valve, the same as the umbrella valve utilized in the Mystfree where almost virtually any air movement will open it and any air movement will close it. The Doherty device allows the crankcase air to escape on the down stroke of the piston, but does not allow any air to enter the breather system on the

upstroke of the pistons, hence decreased crankcase pressure. The Power Vent like the Mystfree differs from any other one-way valve on the market, as the umbrella valve does not lock into its seat. Instead, it floats on its stem.

Decreased crankcase pressure will also assist in lessening oil carryover since there is less air to pick up oil and there will be less interference of displaced air preventing top end oil from returning down the oil return holes.

The decrease in crankcase pressure allows a slight increase in power for two reasons: I quote Tim, “The pistons are not meeting as much resistance on the down stroke since there is less air stored in the crankcase. Testing the Evolutions typically would give us about 3 HP. The Twin Cams would always give us, one possibly two, measurable HP on a Dyno. The second benefit would be better ring stabilization. On the

compression stroke, pressure pushes the pistons down from the top but with significant air storage (i.e. pressure); they also slightly push up from underneath, which causes a ring flutter condition.  Although the vents are only capable of working on a limited basis, their theory really bases on what has become increasingly popular in the automotive racing field, which is the use of mechanically driven vacuum pumps. The results in our experience have been dramatic. For example a 25-30 HP gain based on a 700 HP engine has been

## The 2007 H-D TC96 Scavenging Fix

The new 2007 Twin Cam has a total swept volume of 96 cubic inches or in metric measurement 1584cc (cubic centimeters). The bigger the engines get, the more likely blowby will occur. Therefore, it was only natural that H-D had to do something more to reduce the chances of blowby with its larger displacement 2007 Twin Cams. The Factory felt quite logically that if less oil remained in the engine waiting

to sump back into the oil tank that the rushing displaced air would not be able to mix and pick it up so easily or at least in the same quantities.

H-D has a redesigned and upgraded oil pump. Increasing stroke also means more heat. Therefore, H-D did a good double whammy.

- Harley increases the 2007 oil pump feed to 10% more flow. This ensures better lubrication and cooling since one of oil's major functions is to remove heat.

- On the blowby side, this new pump increases scavenging oil from the bottom end by a whopping 23-percent.

Therefore, we have the potential here for superior results, both in cooling and in controlling a source of blowby, thereby

preventing the new Harley's from marking their spot.

## 124 Safe Cubic Inches

S&S Cycle, Inc. is the venerable performance company that is an industry leader in developing quality parts and engines for the V-Twin crowd.

The 124 cubic inch Hot Set Up Twin Cam using the stock A or B engine casings from 1999 to 2006 is a good example.

S&S also introduces a 124" kit for the 2006-present Dyna and the 2007 to present FLHT's. In order to do this, S&S had to design and make cams with larger journals on either end. Fitment for the rear cam journals requires increasing their size from .875" used on 1999-2006 Twin Cams to a full 1.000" used on 2006 Dynas and all 2007 to present models. The front cam journals also increase in size to fit into the very much

larger parent material bearings in the new cam support plate.

This is deceptive language. The parent material bearings are nothing more than holes cut into the support plate without benefit of the traditional roller and ball bearing combination. They make me nervous but these types of parent bearings have successful use in the automotive industry over a long period-of-time. However, the fly in the ointment is the physics principle of Moment Arm Force. Please see Chapter II: Repairing the Achilles section, Moment Arm Torsional Bearing Wear for a discussion of a possible future timing chest problem. Moment Arm Force may cause the outboard cam journal to oval the new parent material bearing in the cam support plate.

We have built many S&S 116 cubic inch Twin Cams at Heavy Duty Cycles using the stock H-D engine casings with

stellar results. Problems have never been an issue with the 116c.i engines that we built. This demonstrates the quality of the engine and its component parts. One 116 incher has been ridden daily for 6 years now without a rebuild and is still going strong.

- The 116” kit uses a 4 ½” stroke combining with a 4.000” bore.

- The 124” version increases the stroke to 4 ⅝” and the bore to 4 ⅛”.

- This compares with a TC88 stroke of 4.000” with a 3 ¾” bore,

- The TC96 has a stroke of 4 ⅜” and the same bore as the TC88 of 3 ¾”.

- The Evo 80 cubic inch engine has a stroke of 4 ¼” stroke and a smaller 3 ½” bore.

Buy a Crate Engine or Build Your Own?

An advantage to building your own S&S 124 cubic inch Twin Cam using the stock Alpha engine casings or the counterbalanced “B” ones is that the performance mechanic can massage or change components for specific riding styles. This may include examples like cam choice, cam timing, performance ignition, oil pump and tappets or polishing exhaust ports on the heads to name a few. There will also be a cost saving on parts over buying a complete S&S 124-inch engine since the mechanic can reuse parts like the engine casings, oil pump, cam cover, and tappets. It also enables the owner to have an engine with a Harley-Davidson serial number known as a VIN or vehicle identification number.

The crank casings are of superb quality and the best stock ones ever used on a Harley-Davidson. S&S does not supply complete counterbalanced Beta Twin Cam engines although JIMS has 120 cubic inch and 131 cubic inch crate engines available through Harley-Davidson dealers.

Therefore, stock Twin Cam B engine casings are necessary to build the 124" Hot Set Up builder's kit into a Softail compatible counterbalanced stroker.

While S&S provides a one-year warranty for their kit parts that are build purchases while the complete 124" "A" crate engine they supply with their ignition has a 2-year parts and labor warranty if S&S common sense procedures are followed. These types of warranties are previously unheard of. Forever, the only warranty with stroker engines and/or parts has been 50-feet or 50-seconds, whichever comes first.

S&S Cycle, Inc. also provides machining services and/or engine-builds in their LaCrosse, Wisconsin facility.

## S&S 124 Big Inch Options

There are three main options when building a 124” Hot Set Up Kit engine.

### The Poorest Choice

The first, which is a no-go for me, is a kit that utilizes a reverse dome or dished piston to alleviate compression with a stock Twin Cam head. This kit is for use with stock TC88 heads producing a compression ratio of 10.8:1. The dished piston is necessary because the design of the stock combustion

chamber size is for use with 3 ¾” bore for 88c.i, or 3 ⅞” bore for 95c.i not a massive 4 ⅛” bore for 124 inches. The compression would prohibitively be too high if using a flattop or raised dome piston because there is more swept volume forcing into the same size combustion chamber. The only two ways to deal with this is dishing the piston to effectively but inefficiently increase the combustion chamber size or install heads with a larger combustion chamber of about 89cc’s volume. Dishing the piston takes away some of the turbulence, this is necessary to mix the air and gas to obtain a superior flame travel and a higher percentage of burn.

The larger question becomes, why would anyone use a head that flows less than and 80 cubic inch Evo head, which is too small for 88 inches never mind 124 inches like on our build? The stock heads will require much reworking with porting and polishing, the installation of larger valves and performance valve springs. Secondly, there may be starting

issues with this kit since stock Twin Cam heads do not accommodate compression releases.

Part numbers are for black powder coated 124” S&S kits are 90-0133 for TC88A engines, 90-0137 for Beta engines. The kits also come in a natural finish. The reason for this kit is for the economically minded.

A Better Choice

The second kit is a better choice for those that already have the 89cc chamber cylinder heads. They will produce compression of 11:1. This kit does not supply with compression releases but nearly all the 89cc heads will accept them. They will be necessary for problem free starting.

Furthermore, an ignition like the S&S IST Intelligent Spark which retards the ignition automatically just enough to stop

spark knock or pinging when it occurs. This ignition is one of the very best bar none for carbureted engines.

The above kits are 90-0147 for TC88A engines, 90-0149 for Beta engines and 90-0169 for 2006 up Dynas and 2007 FLHT models.

The Best Choice

In my opinion, the third kit offering is only choice for me. However, it costs more than the other two, as it is the most comprehensive offering. The 124” Hot Set Up kit supplies with S&S 91cc Super Stock cylinder heads producing a compression ratio is 10.8:1.

It also supplies with S&S electric compression releases, which are a bolt-on affair through centre holes in the lustrous

and beautiful S&S wafer boxes and separate rocker arm supports.

## Comparing Popular Engine Statistics

The chart compares some popular bores and strokes. Compression may vary depending on pistons used and heads. Where applicable I show the lowest compression ratio. There may be options to raise compression further. The JIMS 120 is really 120.3 cubic inches and the 131 is really 131.5 cubes. The JIMS use of a 4.500” stroke bodes well for reliability, as does their conservative 10:1 compression ratio. To put this in perspective, I find the S&S 124 to be very reliable also with a stroke that is .125” longer and a compression ratio of 10.8:1. The correct ignition with anti-knock (anti-detonation) capability allows for higher torque producing compression ratios.

10.1:1

So, how does the increased stroke on the 124” of 4 ⅝” over the 4 ½”stroke and the increased bore to 4 ⅛” over the 4.000” bore of the very reliable 116” engine impact on the 124 cubic inch engine’s dependability?

Stroke

Since the stock 4.000” TC88 Twin Cam’s stroke is a less than that of 4 ⅝” stroke of the 124, the crank pin is moved out further from to the centre of the rotating flywheels on the bigger engine. As this is a circular motion diameter, the distance is a radius of further from the centre for a ⅝” total increase in stroke.

The 124 cubic inch stroke of 4 ⅝” will have a radius increase of further from the centre of the flywheels for a total ⅛” increase over the S&S 116 incher, the JIMS 120-incher and the 131 cubic inch monsters 4 ½” stroke.

The moving of crank pin location is not an adjustment. We must use different flywheels with the crank pin mounting holes bored closer to the rims of the flywheels when comparing to the TC88 crank. The purpose of increasing the stroke is to obtain a greater swept volume (piston travel), also known as engine displacement.

## The Platform for Bigger Engines

What makes performancing a Twin Cam so easy is that the array of quality parts available gives a multitude of safe

options for going faster. The platform of the Twin Cam engine provides for dramatic and safe increases in displacement through stroking and bigger cylinder bores.

On a stroker, the piston must go further up the bore to a new TDC (top dead center), and back down an equivalent further amount to BDC (bottom dead center). This necessitates other changes. In review, it may require longer barrels so the pistons do not smash into the heads at TDC. Another technique to prevent the piston skirts from smashing into the flywheels at BDC is to use smaller diameter flywheels.

Many times the utilization of a shorter piston is the builder's choice. My main concern is usually the stroke since piston speed increases. There are two increasing concerns here depending on length of stroke.

- The first is piston speed since the piston must do this

travel increase in the same time as a stock stroke.

- Secondly, piston skirt area usually decreases with a stroker piston to make certain it does not hit the valves/head at the top or the flywheels at the bottom of travel.

The piston is usually shorter than a stock one with compensating piston pin relocation. The piston skirt area is the thrust face portion below the rings, front, and rear that makes contact with and supports by the cylinder sleeve in its bore. Furthermore, rear piston thrust surface compromises even more since relieving material is necessary, usually from the rear piston's front skirt to prevent the rear-piston front-skirt hitting the rear-skirt of the front-cylinder-piston at BDC.

Bore

However, on my first 124-inch build a few years ago, my main concern was the bigger bore. Bore in itself is not a problem since piston speed does not increase and it is the safest way to make an engine bigger. Big bore is just as safe as using the stock bore. It even has another major advantage beyond producing more cubic inches; as the bigger bore allows intake and exhaust gases to travel more easily around the valve heads into and out of the combustion chambers.

With standard sized bores, there is sometimes the concern of a valve-shrouding impediment. With tight space, the valves may shroud efficient flow of the air/gas mixture or the spent exhaust gases. Assuredly, there is no valve shrouding with the 124c.i Twin Cam behemoth because of the huge 4 ⅛" bore and even less with the JIMS 131 cubic inch engine with an even larger 4 bore. This is in spite that the valve sizes are increased to 2" on the intake and 1.605" on the exhaust on the S&S Super Stock heads compared to stock Twin Cam valve sizing of about 1.813" and 1.563 " respectively. The bigger

valves allow the increase in flow required to serve this biggest of all production engines for the stock Twin Cam engine casings.

The 91cc combustion chamber heads flow over 30% more air than the stock TC88 heads. They also allow the use of monster .640 lift gear-drive-cams. The heads have larger fins, which provide more cooling area for an increase in heat dissipation.

Performance Valve Springs

The S&S triple valve spring packs compare to an inner and outer double pack used on stock H-D's forever up until 2004 when the factory went to a single beehive spring. The beehive terminology describes the spring winding, which is wider at the bottom than the top in order to thwart spring harmonics

disrupting proper valve movement.

Valve spring tension measures in pounds of seat pressure for installed height and increments to the fully compressed height. The seat pressure will increase as the spring(s) compress from cam lobe rotation dictates.

The reason for installing heavier seat pressure valve springs in an engine is to ensure that the valve train follows the instructions laid out by the more demanding performance cam-lobe-profile. The spring(s) designed is to curb valve float. This occurs when the valve does not return to its seat immediately after the egg-shaped cam lobe rotates to a closing position. High rpms and/or a wild pointed cam profile with all lift relative to duration can easily confuse the valve as the properties of momentum and inertia come into play. The valve will want to keep traveling on its current plane. An increase in duration keeps the valve open longer making it easier for the valve train to follow accurately the now

rounded cam profile.

Valve spring pressure should closely match to an engine's potential rpm range and the profile of the cam. During rebuilds, when performing valve jobs, the installed height will change necessitating shimming of the springs to return them to the desired specifications.

- The triple pack springs are typically 170 lbs at their stock installed height of .1800".

- The pressure of the springs change as the cam lobe opens the valve and compresses the spring pack.

- At .560" lift, the open pressure increases to 390 lbs.

- This increases to 415 lbs at a .630” lift. The reader can see that there is increasing motivation for the valve to return to its seat the more it opens.

Compare this to springs designed for a .585” lift cam that also has an installed height of .1800”.

- The seat force required reduces to 125 lbs.

- Seat force at maximum lowers to 322

Roller Rocker Arms

A cam lift of .640” dictates the use of roller rocker arms. The S&S roller rockers are forged from 4140-steel whereas most others are cast steel, heat-treated and then shot-peened. Shot peening will make surface areas consistent so that there will not be imperfections for potential metal fatigue to begin.

Internal support bushings increase in length to .750” from the stock .500” length used in most rocker arms.

As each cam lobe rotates to its highest position, the hydraulic roller follows its path over the lobe pushing the hydraulic lifter up. This in turn forces the pushrod up causing the rocker arm to rotate pushing down on the top of the valve stem to open the valve. More precisely, it rubs across the top of the valve stem in order to push the valve open. This side rubbing pushes the valve against the guide, which causes wear and resistance because of friction. Anything that can lessen this side thrust is a big bonus in terms of efficiency and horsepower.

Internal engine friction lessens the available horsepower.

Roller rocker arms are superior to stock rubbing rocker

arms. The tip of a roller rocker arm, which actuates the valve opening, is a roller that pushes the valve open with minimal resistance. It rolls across the top of the angled valve as opposed to rubbing across it. As lift increases, so does side loading, friction and inertia, which leads to premature wear and a heavier valve train. Roller rockers along with low friction valve guides act in concert to trick the valve train into thinking and reacting like it is lighter. It is now possible to lower to seat pressure of the valve springs.

This is very important as it relates to valve float with hi-lift cams and/or high rpm riding.

## History of Big Bore Engine Case Cylinder Spigots

I will examine big bore historically using stock H-D engine casings. A major concern will become apparent in building

big-inch engines through bigger bores. The severe limiting factor in utilizing stock H-D engine casings is that there has never been enough meat around the cylinder--spigot-holes that support the cylinders for substantial big-bores. This is critical since the engine cases hold everything together.

Shovel and Pan Big Bore

Big bore using stock Shovel or Panhead engine cases for that matter usually increase bore to 3 ⅝" for very safe results from the stock 3 bore on the 74's and the stock 3 ½" bore on the later Shovelhead 80 cubic inch models. Big Bore can go as high as 3 or even 3 on the Shovel and Pan but the spigot bore of the casings still give best support to the 3 ⅝" cylinders.

The only way to minimally increase the spigot bores safely beyond the 3 ⅝"-inner diameter cylinder liner on pre Twin Cam stock-engine-cases is to add weld around the affected

areas of the spigot bore. The performance mechanic will run into interference problems from the tappet guide blocks on the right side of the engine when adding too much weld. Moreover, the spigot-hole support between the cylinders gets prohibitively weak as the bore enlarges.

Another way to accept the bigger bore cylinders and their pistons is to make the cylinder liner thinner. On a practical street machine, we do not want to do this. Even though this is possible, we do not want to weaken component parts when the stresses of moving larger diameter big-bore pistons further up and down the cylinders via stroking. Stress increases exponentially the bigger the bore and stroke with thinner cylinder liners.

Evo Base Design Big Bore Engine Casings

The reason there are such big cubic engines available present day in the aftermarket, with the Evolution design base the most popular, is the specialty engine casings manufactured specifically for mega bore cylinders. The casings are wider in order to accommodate these huge bores safely and reliably. Some also increase the deck height to allow for longer connecting rods as well as longer skirted pistons that will have more thrust area.

The deck height is the area in which the cylinder spigots are bored. In the above example, the deck height is higher than the stock positioning.

H-D Evo Big Bore

Initial factory 1984-85 Evolution cases were excellent but after that each year or grouping of years were of differing qualities.

There was poor quality aluminum alloy in Evolution 1989 to early 1993 casings resulting in sub par castings. The plethora of problems included cracking engine cases, case porosity allowing oil to seep through into the primary cavity, pulling cylinder studs and machining boondoggles like the sprocket shaft seal hole made too big so the seal floated with oil gushing out around its outer edges. Building a stroker, with its increasing stresses is not a good idea in engine casings such as these. Boring out the barrel spigot holes in these year casings to accommodate the 3 ⅝" big bore barrels leaving a thinner poor material case to support them was risky at best.

Engine casings from 1994 to 1998 were of much better quality.

A major drawback when manufacturing engine casings is

the number of times necessary to clamp them during the machining process. The alloy contorts every time during clamping, which can cause machining distortions as the alloy bounces back to its former shape before clamping. Less clamping means a superior product. As I have said, the Evolution cases require clamping 32-times before fully machined. This is too much by today's standards.

Twin Cam Big Bore

The superb Twin Cam casings are the best ever produced by the Factory. These high-pressure die cast aluminum beauties promote rigidity, strength and durability as the framework for what is inside and on top. The 360 special formulated proprietary alloy's quality is evident as soon as looking at them. The Twin Cam housings require only 5-clampings during production.

H-D uses the modest 4 ⅜” stroke combining with a 3 ⅞” bore to realize 103 cubic inches, a 4.000” bore for 110 cubes and 4.060” bore to attain 113 inches.

Utilizing a relatively conservative 4 ⅝” stroke in Harley Twin Cam engine casings supports the 116” and 124” cubic inch engines. I began to feel very good about the longevity potential for this engine once I satisfied myself that the stock Twin Cam engine casings had sufficient quantity and excellent quality material to support the bigger diameter 4 ⅛” cylinders after boring out the supporting engine spigot holes to accommodate them.

Measuring Spigot hole support material for 124” Cylinder spigots

The limiting factors with big-bore are how thick the support material surrounding the spigot bores is, and how much the mechanic is willing to risk. Tattoo Tony of Heavy Duty Cycles, measures spigot bore support aluminum thickness with a Vernier caliper after boring out these Twin Cam Alpha cases

to accommodate the S&S 4 ⅛” bore cylinders for the 124 cubic inch build. Unlike Evo or Shovelhead engine cases, the Twin Cam has ample spigot support. There is lots of meat after boring case spigots; the right side is .406” thick and the left side is .419 inches. The metallurgy is also much superior to what went before. I really cannot say enough about how good the stock cases are. The final 124-inch specifications are a bore of 4.322” to a depth of 1.545 inches. First, for a 116-inch build, bore the spigot holes to a diameter of 4.075” and approximately .2250” deep. Pay special attention to the specifications as the 116 cubic inch 4 ½” stroke and 4” bore cylinders require a further diameter cut of 4.255” to a depth of 1.550” from the cylinder deck. The 124” engine spigots require a first diameter cut of 4.162 inches also to a depth of .2250”. Stop before hitting the pinion-bearing boss! The cylinder deck is the part of the engine casings that the cylinders sit on separated by the cylinder base gasket. Tattoo Tony is measuring the thickness of part of the cylinder deck in the picture. The 124 cubic inch engine’s 4 ⅝” stroke and 4 ⅛”

bore cylinders need a differing cut depth of 1.545” from the cylinder deck surface along with a bigger diameter cut of 4.322”. It is very important not to exceed the depth cut specification as the spigot hole may break into the top end oil-return-passage in the crankcase to the timing chest compartment. Always refer to the S&S specification instructions. Perform machining to one engine casing cylinder spigot at a time. Debur machined surfaces. After this, there is a weak mating piece of alloy projecting just below the centre case bolt hole (not visible in picture), that the mechanic must remove from each case half.

Machining 124-inch, 4 ⅛” Engine Case Cylinder Spigots

The limiting factors with big-bore are how thick the support material surrounding the spigot bores is, and how much the mechanic is willing to risk. I like a no risk situation, which is with the Twin Cam casings. Bigger bore cylinders usually requires boring the casing spigot holes they fit into to

accommodate the new barrels (cylinders). The Twin Cam casings, on completion of spigot hole boring will have enough quantity and quality case material to support the 4 ⅛" bore cylinders. This is the reason that it is safe to go to 124-cubic inches.

There are two main ways to bore the stock Twin Cam engine casings to accommodate the 4 ⅛" bore cylinders.

- The first way is to use an S&S boring fixture and spacer in conjunction with a boring bar, a precision tool usually used for resizing cylinders for larger pistons.

- The second is to bore the cylinder spigot holes out with a milling machine found in any machine shop.

The final 124-inch specifications are a bore of 4.322" to a depth of 1.545 inches.

- Torque the engine casings together by the 9-case bolts/ studs sequentially in an offset order according to S&S instructions, first at 10 ft-lbs and finally at 15 ft-lbs using the same pattern approach.

- The casing center bolt that locates between the spigot holes replaces with a special S&S case bolt whose mid portion is of lesser diameter for cylinder clearance.

Whenever machining component parts it is best to do so under engine operating conditions so that the parts have normal operational stress. Parts will change shape under normal stress and it is necessary for the finished machining to take on the characteristics of an operating condition.

The milling machine procedure requires the removal of the

eight H-D cylinder case studs before the clamping procedure. However, they are a requirement with the S&S fixture as a hold down where nuts torque on the cylinder studs at 35 to 40 ft-lbs. After spigot boring completion, discard these studs, as they will replace with S&S studs. S&S recommends quite correctly not using studs more than once. It is better to replace them once disassembling the engine because of the torsional stress put onto them during previous usage. The mechanic cannot see the effects of the stress, if any, except in extreme cases. This is a case of better safe than sorry.

The S&S build instructions describe specific boring methods using numerous cuts.

- Briefly, first bore the spigot holes to a diameter of 4.075” and approximately .2250” deep for a 116-inch build.

- The 124” engine spigots require a first diameter cut of 4.162 inches also to a depth of .2250”.

Stop before hitting the pinion-bearing boss!

- Pay special attention to the specifications as the 116 cubic inch 4.500” stroke and 4.000” bore cylinders require a further diameter cut of 4.255” to a depth of 1.550” from the cylinder deck.

The cylinder deck is the part of the engine casings that the cylinders sit on separated by the cylinder base gasket.

- The 124 cubic inch engine’s 4 ⅝” stroke and 4 ⅛” bore cylinders need a differing cut depth of 1.545” from the cylinder deck surface along with a bigger diameter cut of 4.322”.

It is very important not to exceed the depth cut specification as the spigot hole may break into the top end oil-return-passage in the crankcase to the timing chest compartment.

Always refer to the S&S specification instructions. Perform machining to one engine casing cylinder spigot at a time. Debur machined surfaces. After this, there is a weak-mating piece of alloy projecting just below the centre case bolthole that the mechanic must remove from each case half.

## Caveats for a Reliable Big-Inch Twin Cam Engine

I always mention the 5 caveats when I make a statement about the reliability of S&S 116 cubic inch and 124 cubic inch engines in stock Twin Cam casings or for that matter, any motor.

1. The first is the engine must be broken in properly.

2. Second, when broken in, the engine must warm up before grabbing a handful, not by idling but riding very conservatively so there is air-cooling.

3. Third, there are regular oil/filter changes, in fact the more, the better.

4. Fourth, switch to synthetic oil once the engine is broken in such as S&S recommended Mobil 1 20W50 V-Twin synthetic.

5. Finally, use the S&S IST ignition with carbureted models, which is self-timing, has an anti-knock sensor and prevents the great engine destroyer, detonation also known as pinging or spark knock.

An appropriate ignition-fuel modifier is also necessary with EFI models such as the self-tuning Terminal Velocity or the old pro, Power Commander.

## Detonation

The spark plug ignites the fuel mixture. Normal flame propagation generates a heat pressure wave that travels across the combustion chamber. Heat and pressure is the same thing. Heat expands creating pressure. Confined pressure creates heat. If pressure and temperature of the yet unburned mixture reaches an ignition point, the remainder all goes off at once. This is detonation also known as knock, spark knock or pinging. There are many causes for detonation such as poor quality gas, excessive ignition advance, an increase in stroke relative to connecting rod length and valve timing. Anything

that affects relative cylinder pressure will be a factor.

The stroke increase will cause piston acceleration and increase the mean piston speed thus creating heat. An increase in compression also causes heat. In addition, valve timing can keep cranking compression high.

Cylinder Stud Installation

Double-nutting two threaded nuts onto the exposed cylinder-head stud threads allows for removal of the H-D studs. Tighten one nut against the other as a lock; then turn the lower locking nut counter clockwise with a wrench to remove the stud portion threaded into the engine casing.

- Examine the case threads carefully for burrs or defects.

- Gingerly thread the new S&S studs into the casings and

remove.

- Inspect, once again for burrs and defects. If there is a burr or defect, the thread might compromise. This will certainly affect the torque rating and therefore cylinder seal to the crankcase.

- Apply the supplied red 262 Loctite onto the stud before insertion.

- Use the double-nutting procedure except turn clockwise this time to a torque of 10 ft-lbs.

- Repeat this process individually for the remaining seven cylinder studs.

## Piston Oilers

Now install the supplied S&S piston oilers that squirt oil up into the undersides of the pistons. This will remove in excess of F of heat from the piston domes through oil cooling properties. Then, install the stock H-D pinion bearing into the right side case pinion race.

## Cam Bearings

Both S&S Cycle, Inc. and I highly recommend tossing the stock INA rear cam journal bearings on all engines preceding the 2006 Dyna and replace with the tried and true Torrington B148's and using Torrington B168's in the later Twin Cams.

## Frame Clearancing

After the machining procedures, a thorough cleaning and

cylinder stud installation, it is time to mock up the engine with gaskets that affect engine height to double check clearance with the frame and gas tank(s). It is especially important to ensure sufficient clearancing for Softail models. Rubber mounted TC88 frames rarely present any clearancing troubles.

- There must be at least .060” clearance between the engine and gas tank(s) for rigid mounted Softail.

- Rubber mounted FLHT Bagger and Dyna engines and their respective gas tanks require .250” clearance.

The 124 inchers do not usually have fitment issues.

Tappets and Limiter Kits

S&S provides two different limiter kits depending on the year of the engine build.

- Earlier 1986-2002 H-D hydraulic lifters (18538-99, 18538-99A) require S&S # 33-5339 HL2T limiter kit.

- Later 2003 to present Twin Cam lifters (18538-99B) require the S&S # 33-5338 HL2T limiter kit.

One limiting spacer installs in each hydraulic lifter. The spacer design limits the range of travel of the hydraulic portion of the lifter. Installation and adjustment of either HL2T limiter is the same.

1. Remove the retaining circlip from the top of the stock (tappet) hydraulic lifter and then remove the piston assembly and spring from the lifter body.

2. Install one of the four limiting rings for the year lifter in use into the hydraulic body bore.

3. Reassemble the stock parts in the order of removal.

The limiter-hydraulics requires the use of adjustable pushrods provided with the Hot Setup Sidewinder 116” or 124” Kit.

1. Install the pushrods loosely and rotate the engine until one hydraulic is at its lowest point.

2. Collapse the hydraulic by adjusting the pushrod adjuster down until it bottoms against the travel limiter.

3. Adjust the pushrod up (shorter), by one full turn. This is equivalent to 6-flats of the adjuster nut.

4. Tighten the pushrod locknut without altering the adjustment.

5. Repeat this procedure for the other three pushrods.

Evo and Twin Cam S&S Sidewinder engines provide limiter rings as part of the kit.

The advent of Evolution engines in 1984 saw the deficient Shovelhead hydraulic lifters replaced with the very much stronger and supportive automotive types in use to present day. Twin Cams improve their effectiveness further by reducing pushrod angles as they extend up to the rocker arms from the hydraulic lifters. The two-cam system means that cam lobes are almost directly under the specific valves they

command.

The function of hydraulic tappets is to reduce noise as the valve train expands and contracts. More importantly, they close up space between mating parts to maintain correct specifications such as cam lift and cam timing as the valve train contracts when cool and expands when hot.

Solid lifters were mandatory in pre-Evo performance engines because of exacerbated pushrod to rocker arm valve angles and their inherent weakness under duress. The older lifters would also collapse if a grain of dirt debris settled between the hydraulic ball bearing and its seat. Debris collapsing lifters was common because the higher wear characteristics of yesteryear engines and their substandard oil filtering systems relative to the present.

- Shovel filters do not have an accurate micron (millionths of a meter) rating for oil particle separation and removal.

- Evo filters have a 30-micron rating.

- The first Twin Cams reduce debris capture to 10-microns.

- Recent Twin Cams halve this particle separation to 5-millionths of a meter (microns).

Solid lifters are not necessary in S&S big-inch Twin Cams because of their superior hydraulic lifters, straighter in-line pushrods, and limiting kits. Limiter kits prevent the hydraulic lifters from collapsing or bleeding down under duress. Cam timing and valve timing specifications remain constant, as they should because the limiters keep the hydraulics pumped up. In effect, limiters allow for hydraulic quietness and solid

performance. It is not often we get the best of both worlds without compromise.

In my opinion, these limiters are a must in performanced engines with stronger valve springs and with performance cams that have more aggressive cam lobe profiles although I have found some cam profiles like Wood Performance ones that run better without the limiter kits.

These kits also work well in stock engines with aggressive riders. The roller tappets that follow the cam lobe profile have physical difficulties dictated by the principles of inertia and momentum when engine operation is assertive at high rpms. In other words, following the cam lobe religiously becomes more difficult especially at the top as direction changes from up to down. This can affect and alter cam timing. Exacerbating the situation is a tendency for the hydraulic

lifters to partially collapse unless installing limiters in these tough situations further complicating the issue.

Higher compression is also an aggravating factor. The installed limiter ring will not allow the hydraulic to collapse or bleed down fulfilling the same function as a solid lifter when necessary.

The rings have the added blessing of preventing bleed down of the hydraulic lifter at rest. Cranking compression increases when lifters bleed down. This sometimes causes difficult starts. After the engine fires, oil will pump back into the lifters making them hydraulically functional and thus quiet.

If using limiters in a stock bike, it is necessary to replace the stock non-adjustable fixed length pushrods with adjustable

pushrods in order to obtain the proper plunger positioning of the hydraulic lifters with installed limiter rings.

The complete crate 124” S&S Twin Cam engines use S&S high performance hydraulic tappets (# 33-5351) fitted with their HL2T limiter rings.

Engine Covers

The S&S Cycle, Inc. engines accept the stock Twin Cam timing covers unless the builder wants the aesthetics of another style or the S&S billet cover.

All Twin Cam engine covers, namely the cam cover, the split two-piece rocker wafers and the guide block covers are just that…covers. Predecessor H-D engine covers are dual function since they also act as load bearing supports for other engine parts. Sometimes, oil leaks are difficult to control

because of vibration and the movement of the cover load-bearing parts. The support role also assists in the transfer of noise to the outside world, an emissions faux pas.

Compression Releases

Big inch engines have had a bad reputation in the past for starting because the compression increase overwhelms the starter motor and battery capacity.

The traditional way of combating this is through long duration and high overlap cams.

Therefore, a valve will normally begin to open just before the end of the previous cycle and close after its theoretical cycle. This is also necessary for the lag time it takes combustion to occur after spark ignition. If a valve begins to

open 20-degrees crank rotation before its cycle and closes 35-degrees after its cycle then it will have duration of (20 + 180 + 35) Intake and exhaust valves will have different opening and closing figures because of their differing functions.

The S&S 124” uses monster .640” lift gear-drive cams with duration on the intake valve and on the exhaust.

- The intake valve with opening closing figures of + + = duration) indicates that the valve will open BTDC (before piston top-dead-center), and close at ABDC (after piston bottom-dead-center) of its cycle.

- The exhaust valve with opening closing figures of + + = duration) indicates that the valve will open BBDC (before piston bottom-dead-center), and close at ATDC (after piston top-dead-center).

Since cranking compression begins at bottom-dead-center of the cycle, effective compression will not start until later because the valves on the adjacent cycles are open longer. The longer the duration of the cam the less time the upward piston has to develop cranking compression and thwart the starting system.

Cam Comparison Chart

“G” stands for gear drive as opposed to stock chain drive. I use the bolt in, mild but performances effective 510G S&S gear drive cams as comparison with the H-D stock profile and the wild 640G’s. Emissions control cams sometimes use negative valve duration opening and/or closing times to control pollutants. This practice is counterproductive for performance purposes that acknowledge the necessity to

compensate for mechanical parts movement and spark ignition lag times. Advance and retard spark ignition compensates for piston speed and other factors as well as taking advantage of scavenging for an increase in both cylinder fill and purging.

Many mechanics will add higher torque starter motors; thicker cables and higher cranking amp batteries but these are not necessary. The starter motors and batteries in the Twin Cam are the best quality ever supplied with a Harley. Nonetheless, it is paramount that they are in excellent operational order.

However, if the heads on a high compression motorcycle have no space to install compression releases or releases are lacking, then longer duration cams with upgraded starter, battery and cable system will be necessary with troublesome

motors.

The 124” kit comes with both electric compression releases and compatible heads and rocker box assemblies. They also look trick and are aesthetically pleasing. A compression release opens during the starting procedure, allowing some of the building cylinder pressure to escape. A small valve in the compression release electrically opens when starting. The compression-release-solenoid activates the valve when pushing the starter button. The reduction in cylinder pressure during starting allows the engine to turn over easily and thus fire. Once started, the valve closes automatically and the engine produces full power with full cranking compression.

S&S also has manual compression releases that activate by a choke like control cable. The rider must pull the cable when starting the engine. The compression release automatically shuts off when the engine fires returning to full compression when running. The manual (90-4925) kit can install with any

type rocker box but the electric ones (90-4915), require a central hole or chimney in the center of the box in order to house the compression release solenoid.

The trick die-cast boxeswith central chimney holes come in the 124” kit. The 2-piece rocker boxes (chrome 90-4110, polished 90-4105, or unfinished 90-4115) use leak resistant Viton O-rings.

Rocker Valve Covers

Something every mechanic will appreciate is that the engine does not need removal when taking off or putting on the S&S rocker covers if stock or close to stock. However, taller cylinders on big-inch engines will require removal when servicing the covers. The rocker boxes will accept up to .640 lift cams when using S&S valve springs, which is huge when

comparing to a stock TC88 cam lift of .473” or 2007 TC96 lifts of .471/.474”before any clearancing is necessary. However, as always, check this tolerance as with any clearance in the engine. It is incumbent on the builder to check religiously all tolerances all the

The S&S rocker boxes use an exclusive one-way valve built into the S&S rocker cover gasket to control oil carryover in head breathing engines.

The Hot Set Up Kits also come with the aforementioned parts like their forged roller rocker arms, completely setup heads matching to the S&S .640 lift gear drive cams along with the aforementioned 4 ⅝” assembled stroker flywheel set, 4 ⅛” cylinders and pistons, adjustable pushrods, hardware like cylinder studs, pushrod tunnels and more plus all gaskets.

The above 124” kits for TC88A engines are part #

90-0132, for Beta engines #90-0136 and for 2006 up Dynas and 2007 FLHT models #90-0171.

An ignition, carburetor, and exhaust will complete our 124” build. The choice of all three is extremely important to preserve and increase power generating by the engine.

Cam Timing

Cam timing on stock Twin Cams can vary by 4-degrees and this is a direct result of the loose fit of the stock cam sprockets. One only has to look at these sprockets to see that exact timing is impossible. Examining the new 2006 Dyna and 2007 to present Twin Cam chain sprockets will demonstrate that nothing has changed. Furthermore, if the stock 1999-2006 cam chain spring-loaded-tensioners are not strong enough to dictate the tension of the timing chains then cam

timing can and will vary beyond the 4-degrees mentioned above.

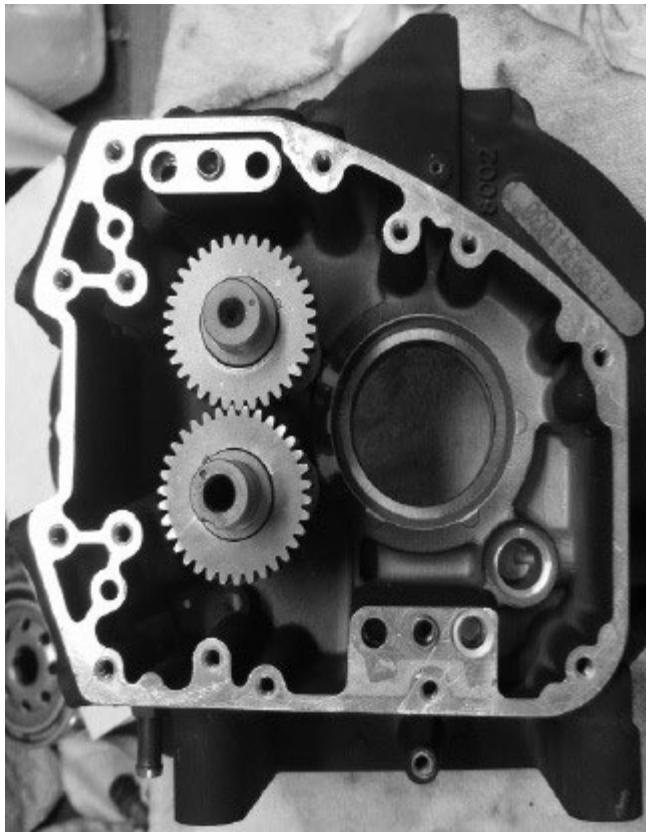
It appears that the hydraulic chain adjusters on the 2006 Dyna and all 2007 to present Twin Cams will work better than their predecessor spring-loaded tensioners.

How far timing will vary depends on the seat pressure of the valve springs, the lobe profile of the cams and the rpm range. Seat pressure must match to the cam profile in order to prevent valve float or its cousin valve bounce. However, if seat pressure is more than necessary, then the valve train will effectively be heavier causing unnecessary stress on parts.

Bigger cubic inch engines require modified cams to service the air demands of the increase in swept volume of the cylinders. Higher cam lifts very much need stronger valve

springs to ensure the valves follow the dictates of the rising and lowering camshaft lobes. In my opinion, cam timing will vary by up to another 4-degrees with strong valve springs necessary for .640 lift cams if retaining the earlier stock chain drive timing chain system. The jury is still out on the later timing-chain hydraulic adjusters although they are definitely superior. We need to use gear driven cams in all Twin Cam engines to keep the cam timing precise.

Big Inch Cam Clearancing

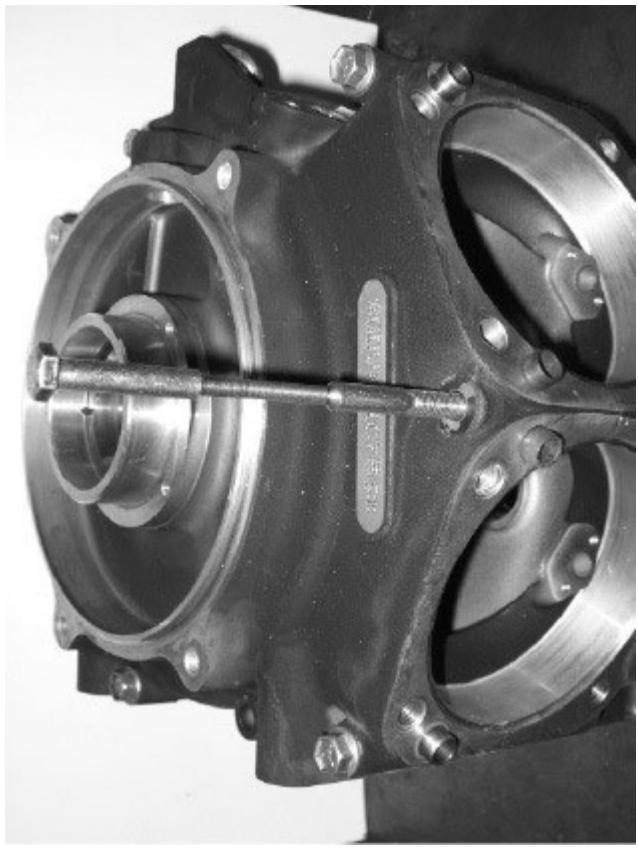


Pictured is the right Twin Cam engine casing with S&S .640” lift cams. The drive gears are separate from the cam journals but press fit onto the journals via keyways. The timing mark on each drive gear face is visible. They must align with each other when rotated to this position. Below the cams is the pinion race. The massive pinion roller bearing sits in this race and supports the pinion shaft as it extends from the bottom end compartment into the timing chest compartment shown in this picture. To the right but on the outside engine casing is a raised casting number 8002. The indented stamped number just below the casting number on an angle on a machined case surface is the engine serial number also known as the vehicle identification number (VIN). Clearancing the cams begin with using the right side engine case separate from the rest of the engine. Install the cams into their right case rear cam journal bearings. Turn them at least 360-degrees to observe any clearance situations between the cam lobes and the crankcase. Thoroughly check for two areas of concern. The pinion-bearing boss on the inner case perimeter that supports the

pinion bearing and the crank pinion shaft that goes through the bearing centre is the first. Cams of this size will invariably have cam lobe interference with this boss particularly the front cam's rear intake lobe that is barely visible in the picture. The second is cam lobe interference with the two tappet guide bosses, above the cams, each supporting two hydraulic tappets on the outer perimeter. There must be a minimum of .030" (30-thousandths of an inch), clearance between the affected cam lobes and these right-casing outcrops. The clearance achieves through either grinding or machining only the material necessary, as I do not want to weaken any engine case castings. S&S has already done the other main clearancing function during cam manufacturing although it is always imperative to check. Any interference between the .640 lift cam lobes eliminates because the gear drive reverses the rotational direction of the rear cam. The mating cam lobes on the two stock TC88 .473 valve lift and early TC96 .471/.474" valve lift rotating cams come close to each other for an instant every 360-degrees. Add too much lift

and they will collide with disastrous results. The S&S gear drive front cam lobes are pointing in the same direction with variances for performance timing as the stock front cam. The rear S&S cam lobes point in a different direction than the stock rear cam. Taking the gear driven reversal into account, the valves will still open and close at the right times. With cam timing, lift and duration variances designed to produce power; the reverse lobes open and close the valves in the same order as the chain driven TC88or TC96 cams.

124” Center Case Bolt-Cylinder Interference



Engine casings clamp together with three types of hardware. A stud may protrude through both casings with flat washers and nuts on either threaded end. Secondly, a bolt may protrude though both casings fastened by a flat washer and nut at the other end. The other visible engine case bolts in the picture are of this type. The third way is for a traditional bolt to slide through the right or left side engine casing and thread into the opposing case. The center case bolt on a Twin Cam between the cylinders does not go right through the right engine casing but threads into it from the left engine casing in order to clamp the case halves together. With smaller big-bore applications, enlarging the spigot holes may also cut into this center case bolt. S&S chooses not to risk damaging the bolt's integrity known as the tensile strength of the bolt in mechanical language. The casing center bolt that locates between the spigot holes replaces with a special S&S case bolt whose mid portion is of lesser diameter for cylinder clearance. Pictured is this clearancing center bolt sitting at the edge of the center case bolthole in the right casing. Axtell Sales

(www.axtellsales.com) uses a different methodology. Their philosophy is as follows. If boring the spigots with the top stock case bolt installed, the boring tool will cut into the bolt. This can result in a loss of tensile strength of the bolt, which may allow a possible oil leak. If re-installing this machined case bolt incorrectly, damage may occur to the cylinder flange and mating piston. The stock Twin Cam casing top bolt, which is the center bolt between the cylinders bolts through the left casing into a threaded portion of the right casing. In other words, the bolt does not go right through both case halves. Axtell recommends eliminating this system in favor of the case bolt going through both case halves for superior and more consistent clamping. Therefore, before boring the cylinder case spigots, remove the piston oilers from inside the right case spigots. Drill the right case top partial bolthole through with a ¼" drill bit, using a drill press or milling machine. Furthermore, spot face the outside of the cases around this hole to allow a flat sealing surface. Now, reassemble the engine casings following the H-D manual

specifications except for the top hole. Install the new ¼” bolt and nut with the bolt head on the right side case and the nut on the left one. Torque this top bolt to 85 in-lbs not ft-lbs. If machining the top case bolt, instead of following the above procedure, ensure the bolt positions correctly through the casing halves so as not to obstruct the cylinder spigots. If there is incorrect positioning, the bolt will distort the cylinder flange spigots causing grievous harm to piston and cylinder alike. Pictured on the cylinder spigot deck are the four hollow locating-dowel-pins for the cylinders alignment, two for each cylinder. The outer dowel pin on either spigot serves double duty as an oil-return router for oil returning from the top end down through the head into the cylinder. These oil-return dowels funnel the return oil from the cylinder and transfer it into the engine case oil-galley that travels over to the timing chest. The other six visible cylinder deck holes are for threading in six of the eight-cylinder tie down studs. Behind the center bolt is the VIN or vehicle identification number for the engine. Behind this is the engine case-sealing ring that the

inner primary casing fits around with a sealing O-ring. The center hole is for the sprocket shaft bearing(s) that support the power-side sprocket shaft as it protrudes into the primary casing. These Twin Cam casings are 1999-2002 because they accept the very reliable OEM 9028 opposing and tapered Timken bearings and races in use in various configurations from circa 1956 to 2002. The alternator stator fits into the area between the primary sealing ring and the sprocket shaft-bearing insert.

## Big Inch Cam Clearancing

I also need to address a few clearancing problems with cams that have so much lift. It is best to check for clearancing while the engine is apart for the stroker-big bore build.

With the crankcase halves separate,

1. Install the cams into their right case bearings.

2. Turn them at least 360-degrees to observe any clearance situations between the cam lobes and the crankcase.

3. Thoroughly check for two areas of concern.

4. The pinion-bearing boss on the inner case perimeter that supports the pinion bearing and the crank pinion shaft that goes through the bearing centre is the first.

5. The second is the two tappet guide bosses, each supporting two hydraulic tappets on the outer perimeter.

6. There must be a minimum of .030” (30-thousandths of an inch), clearance between the affected cam lobes and these

right-casing outcrops.

The clearance achieves through either grinding or machining only the material necessary, as I do not want to weaken any engine case castings. S&S has already done the other main clearancing function such as ensuring there is no cam lobe interference problems during both cams 360-degree rotation during cam manufacturing although it is always imperative to check.

Sprocket Shaft Bearings

As I discussed, all Harley's from circa 1955 to 2002, which include late Pans, all Shovels and Evolutions plus early Twin Cams use a fixed and opposed tapered Timken spaced bearing kits (2 races, 2 bearings; spacer kit). This kit determines and ensures accurate and reliable flywheel crank endplay. The tapered bearing races fix into position as they press into the

left engine casing with the tapers facing outwards opposite each other and further held in place via beefy circlips.

1. One Timken bearing, the taper facing out presses onto the left crank's sprocket shaft also known as the power or power output shaft.

2. The crank then installs into the case so the tapered bearing fits into the tapered case race.

3. The other Timken bearing will then press onto the protruding sprocket shaft, taper in, until it fits into its companion race.

4. A precise spacer separates the bearings and races whose differing size thickness vary in thousandths of an inch to determine endplay of the flywheels inside the engine casings.

5. The left engine casing now mates with the flywheel assembly.

Endplay is extremely important. If allowing the flywheels to float back and forth on their supporting bearings in both left (sprocket shaft power side), and right (pinion side) casings, the crank assembly could easily walk and hit either of the inside casings.

The tapered Timken kits worked so well for 47 years, I am one of many in surprise when H-D saw fit to replace the Timken's with a roller bearing in 2002. At Heavy Duty Cycles, I have found a number of 2003 and later Twin Cams where the flywheels have begun to walk beyond their mandated endplay specifications with unpleasant consequences. Please see the section, 2003-Present Twin Cam Engine Casing Solution earlier in this chapter for resolution of this potential

problem if uncomfortable with the sprocket shaft roller bearing.

## Internal Engine Clearancing

The performance mechanic can check some of the internal clearancing situations and perform them during engine assembly. However, do the next four checks with the flywheel assembly sitting in the right hand engine casing, pinion bearing installed.

- Set up the pistons with wrist pins but without rings or wrist pin clips onto their respective connecting rod small ends.

- The pistons have directional fitment and must install according to the instruction sheets.

- The front of the rear piston thrust surface skirt has a notch cut in the bottom. Each cylinder carefully assembles over its respective piston.

- Hold the cylinders in place with one nut each.

- First, rotate the engine carefully so the pistons are at top dead center (TDC).

- The piston deck should be flush with the cylinder head gasket surface.

1. The final assembly gasket thickness will provide the 0.45” squish required for this particular engine. This becomes the piston to head clearance. Squish refers to the forcing of the turbulently mixing air and gas from the swept volume of the cylinder bore into the smaller combustion chamber. It is

crucial that the piston does not rise above the head gasket surface as parts may interfere with each other. This is a problem for S&S to deal with if it occurs.

Next, ensure proper clearance of the turning connecting rods to the crankcase.

The Harley-Davidson Twin Cam crank pin at 1.671” diameter is a mammoth size. This compares with the smaller but reliable diameter of 1.250” in the Evolution and its predecessor engines. However, in big inch strokers, the massive crank pin size interferes with clearance as the engine rotates. The increase in angle that the connecting rods must negotiate especially at the nine, six, and 3-o’clock positions will create clearancing problems between the male and female rods as well as other areas. Therefore, S&S uses a 1.500” diameter crank pin, which is still very much bigger than the

Evo. Furthermore, the stock Twin Cam connecting rods do not incorporate con-rod-races in order to make them rebuildable. The other reason for S&S using a 1.500” crankpin is to allow room for connecting rod races. Ergo, the S&S connecting rods are rebuildable rather than disposable. Aftermarket Evo engines are currently commercially produced up to 131 cubic inches using the 1.250” crank pin. S&S’s experience with nitro burning drag race engines demonstrates that this 1.500” diameter is more than strong enough since nothing is tougher on an engine than nitro fuel and the mega cubic inches, this fuel serves.

2. Rotate the flywheels carefully once again until the connecting rods are closest to the engine casing and cylinder spigots, both in the front and the rear. There must be a minimum clearance of .060” at their closest point between the connecting rods and the crankcases and/or cylinder spigots.

Do not remove material from the con rods! Grind or machine the crankcase halves or spigot(s) until there is .060” clearance after covering or taping the pinion bearing, cam bearings and the piston oilers. This, however, is usually a problem the mechanic will encounter with longer strokes or beefier con rods. As always, thorough cleaning is the only acceptable cleaning procedure.

3. Now remove the cylinder hold down nuts and move each cylinder up their respective studs about one-half inch. Insert wood or plastic between the cylinders and cylinder decks to keep them in position. Rotate the flywheels gingerly, once more until the pistons are at BDC (bottom dead center) where they are closest to each other. The standard clearance of .060” is necessary at the pistons closest point to each other.

4. If clearancing is necessary, disassemble the parts and carefully file off the piston skirt(s) until we get our .060” green light.

5. Weigh both pistons after the next and final step to see that they are the same. Remove material from under the piston boss from the heavier piston.

6. Check the last clearance between piston skirt and the piston oiler in each cylinder while the pistons are again at BDC.

7. The clearance is again .060” for each cylinder piston and oiler. If under, notch the offending piston(s) skirt with a file until there is the mandated minimum clearance; progressively smooth ground areas with finer and finer sanding cloth.

8. Ensure pistons weight the same. If not, follow the procedure in step 4.

We can now disassemble our clearance test parts, clean and begin final bottom-end-engine-assembly according to H-D approved specifications.

These occasionally over-ride by S&S procedures outlined in the instruction sheets such as the torquing of the case bolts first to 10 ft-lbs and then 15 ft-lbs sequentially in the pattern specified.

Cam installation is very different from H-D specifications. This is due to the increases in lift, duration, and overlap.

## Top End Clearance Verification

Verify top end clearancing specification tolerances after installing the pistons and wrist pins in their mating cylinders on the bottom end.

1. Paint the piston intake and exhaust valve pockets with machinist’s blue.

This is a harmless coloring agent.

2. Secure both valves in one head, against their respective valve seats with an elastic band without valve springs.

3. Assemble the head to its cylinder without a head gasket and fix snugly with one head bolt.

4. Lower each valve individually and rotate it as the valve head touches the top of the piston to mark the machinist’s blue.

5. Remove the head and inspect the contact area of the valve heads to the piston valve pockets.

6. First, the valve should fit into its mating valve pocket cutout in the piston dome. There should be minimum .060” clearance around the periphery of the valve head in the pocket.

7. If not, disassemble, grind or machine the valve pocket in the piston dome until there is sufficient clearance with the valve sitting flush in the pocket.

8. Repeat this with the other three valves and their mating valve pockets.

9. Next, put some mechanics putty (Plastocene) into all four-valve pockets with the pistons and wrist pins without rings in their cylinder bores to check the valve to piston

clearance at TDC.

10. Install the assembled cylinder heads and head gaskets onto the cylinders with the rocker covers, roller rockers, and pushrods without pushrod tunnels.

11. Adjust the pushrods down collapsing the hydraulic lifters until they lightly contact the limiter rings inside.

12. Rotate the engine in its normal direction carefully. If meeting resistance, stop immediately and disassemble. Rotate two complete revolutions so that all four cycles or strokes move through their full rotation.

13. Disassemble and measure the putty in each valve pocket. Intake pockets should have minimum .060” clearance but the exhausts need .080” clearance between the valve face

and the valve-pocket-recess face. Grind or machine pocket recesses if necessary to obtain minimum clearance.

14. With most engines, it is now necessary to check valve-to-valve clearance in each head as they overlap at their closest point to ensure .060” clearance but the bore is so big on the 124” that this is not necessary with this build.

15. Once again, weigh both pistons after to ensure that they are the same. Remove material from under the piston boss from the heavier piston.

I am now ready for final engine assembly according to H-D approved specifications except when S&S procedures override them as outlined in the supplied instruction sheets.

Cleanliness and Lubrication

Check and clean all internal oil passageways with appropriate sized drill bits relative to passage size. Use compressed air to remove dirt and filings. Wash all separate parts in the final assembly with soap and water. Now re-clean the internal engine parts and gasket surfaces with a cleaning solvent that does not leave a residue. It is necessary to prevent oxidation such as rust by coating all the iron or steel parts with a film of oil. Cylinder liners are especially prone to immediate rusting with thoroughly cleaning.

Your hands and particularly your fingertips have a natural oily but corrosive agent film on them. Wash your hands with soap and water before and during engine assembly. Avoid touching bearing surfaces, cam lobes, journal ends, and internal mating steel parts as much as possible. This is initially awkward but time and experience will make this second nature.

Recoat all parts and mating surfaces with high quality engine oil, the same as will run in your engine.

Finally check gasket surfaces for flatness especially the head gasket area on both the cylinder and the head using a machinist's flat or straight edge. If an area is not flat, disassemble and machine to specification. If affecting compression or other important tolerances, compensation is now necessary. For example, if planing one surface of a mating cylinder or head, the counterpart on the opposite head or cylinder should also receive this planing procedure.

This may affect valve to piston tolerances. Recheck all affected tolerance specifications.

To save time and uneeded work, I check gasket surfaces for true at the beginning of the build and replane if necessary.

# Big Inch Engine Breathing

The increase in the engine's lung capacity for air increases 30% from 88 cubes to 124 cubic inches in this particular case, a 1999 carbureted Dyna. Power is all about air, its delivery, and removal. Gas cannot burn without correct proportions of air. Air delivers to the combustion chamber initially by the vacuum creation of the descending piston on the intake stroke.

The further packing in of air occurs especially at higher rpm's by the scavenging forces of exhausting spent gases out the exhaust valve during intake/exhaust valve overlap. This initiates a Hoover effect sucking in a higher percentage of intake charge.

The engine needs to pressurize the air and gas mixture first by the rising piston on the compression stroke before ignition. Burning fuel creates heat expansion and this pressure drives the piston down on the power stroke, turning the crank, which ultimately turns the rear wheel.

The engine now needs to expel the burned remnants out the exhaust valve, tract, and exhaust system by the rising piston so that the 4-cycles of the Harley 4-stroke engine may begin to repeat the cycles.

124-Inch Carburetor

A big carburetor for a big engine, the S&S Super G, accomplishes intake on this particular 124-inch Twin Cam. For this build on a 1999 Dyna, the purchase of the G carburetor is necessary for the set up design for 1999-2001 Twin Cam 124” engines. The G carburetor kit comes with the

correct manifold and a Desperado air cleaner cover.

The increase in cubic inches demand a better breathing tract and the air cleaner cover and filter is part of that. The aesthetically pleasing and traditional Tear Drop S&S air cleaner is slotted to allow more air to access to the wider synthetic media, pleated air filter element.

The mechanic may also use a 48mm Mikuni Smoothbore carburetor.

If using EFI, there is a choice of S&S single bore EFI induction kits or dual bore VFI (variable fuel injection), with optional wild looking and functional tuned induction kits.

Exhaust

The S&S slip-on mufflers provide great midrange torque but will not develop maximum peak horsepower for large displacement engines like the S&S 124c.i T-series, Twin Cam A (Dyna or FLHT models). The same applies whether the engine it is a complete engine built by S&S or one of their dealers such as Heavy Duty Cycles (www.heavydutycycles.com), or a Hot Setup 124 kit installation in Twin Cam A or B (Softail models) Harley-Davidson engine casings.

To achieve more high rpm horsepower with engines like the 124”, the use of another aftermarket exhaust system like a two into one collector system designed for big inches. The collector muffler will support a bigger volume of exhaust gases and have a smaller restrictor pipe exiting the muffler usually equal in diameter to the head pipe.

The header pipes will likely be a larger diameter than the stock 1 ¾" and may be a different length than one designed for a smaller engine. Size and length of header pipes along with collector backpressure will help determine the position of the powerband. The stock 1 ¾" header diameter will create more low-end torque whereas a wider diameter will move the torque band up but increase top end horsepower. A bigger collector is also desirable.

Exhaust technology is beyond the realm of many mechanics and involves complicated physics principles. If the reader wants a guide to what works best, just observe the exhausts on Japanese and Euro super bikes or those found on the very fast Buell motorcycles. Please see Chapter VIII: Eliminating the Performance sections, The Exhaust and Performance and Power Equals and The Catalytic and Exhaust and Mufflers Silencing and and Exhaust Pipe and Exhaust Effect on Fuel and Cam Overlap, Torque, Cylinder Fill, and

Exhaust.

## Engine Installation

The following point seems logical but needs saying as good builders will spend much time and money building an engine in sterile conditions.

However, some violate this principle of cleanliness when installing the engine in the chassis and firing it up. All will put fresh oil in the tank but some neglect thoroughly to clean out the oil tank and oil cooler before doing so! There are those that also forget to throw away the old flexible rubber oil lines and put new ones on the bike to ensure cleanliness in the whole system. Reuse solid metal Softail oil lines after flushing and cleaning in the same way as the oil tank and cooler.

Finally yet importantly, tune the Harley from front to rear for two reasons.

1. The first is safety. The very much faster motorcycle must now handle and stop to the best of its tuned ability.

Chassis alignment along with servicing wheel bearings, steering head bearings and rear fork bearings or rubbers as the case may be is prudent if not essential. Properly inflate tires of good quality with lots of tread remaining separate the rider from the asphalt. Servicing the hydraulic forks with the correct prescribed amount of clean oil for dampening purposes is next. If using an H-D Springer, check all component parts and ensure fork legs are straight as they do bend. Upgrade rear shocks and adjust for rider and motorcycle weight. Belts and chains also need correct adjustment,

alignment, and maintenance. Brake pads and rotors must be within specifications.

2. The second reason is to avoid wasting performance in transferring all that new power back to the rear wheel.

Many of the above tuning steps also assist in power transfer.

Performance Clutch

An improperly adjusted clutch defeats the purpose of building the engine. In fact, why use a stock clutch or diaphragm spring at all? If there was ever a time for upgrading the power transfer capabilities of the clutch this is it!

At the very least put in an upgraded S.E., Alto or Barnett clutch and heavier diaphragm tension spring.

- The serious will install clutch packs such as the Barnett Scorpion (607-30; 1999-2006 or 608-30; 2006 Dyna and 2007 to present all models).

- Other examples are an S&S 1999-2006 performance clutch (56-5150, 36-tooth clutch hub sprocket/cable, 56-5151, 37-tooth hub sprocket/cable 56-5152, 36-tooth hub sprocket/ hydraulic or 56-5153, 37-tooth hub sprocket/hydraulic)

- Or a S.E. Race Clutch Kit (37970-03A, cable only) as a sensible upgrade.

The above clutches except where I note, will mostly fit 1990 to 2006 excepting the 2006 Dynas. The 2006 Dyna and

all 2007 to present Big Twins have radical changes to their primary components along with a 34-tooth engine sprocket and a 46-tooth clutch sprocket. See your bike shop and trusted mechanic for new clutches coming online.

Many of the 3”-primary belt drive assemblies with original use for racing purposes are now available for street use. Many include excellent clutch packs. Please see Chapter X: The Power section, Performance Clutching for much more information.

Gear ratio selection along with a 6-speed transmission will help deliver the power band suited to both the big 124” engine, motorcycle type whether it be Softail FLHT Bagger touring or a Dyna rider weight and riding style. Please see Chapter X: The Power section, Gear Ratios Secondary Gear and Final Drive Gear and Internal Transmission Gear and USA

Overall Gear and Performance and Comparing and Torque Transfer and and The S&S Helical 6-Speed for in-depth discussion on these topics. This subject is one for the knowledgeable who can tune the motorcycle's power delivery to satisfy the wants and needs of both rider and a big-inch power plant.

## Warranty

When obtaining quotes for your particular build always make sure, the lower price has not left out the support parts issue. While we are at it, ensure your mechanic works in a reputable longstanding shop that has insurance and is not living hand to mouth.

Warranty is invalid with a bankruptcy or midnight move. Every shop has a reputation. Ensure yours has a reputation for reliability and backing up their warranty issues.

Sometimes the more expensive shop is including labor and parts replacement that another is not. Heavy Duty Cycles includes bearing and bushing replacement not included in the kits with big inch builds leaving nothing to chance.

S&S has a very fair warranty system. In my experience, they have bent over backwards with legitimate concerns and always give the owner the benefit of the doubt. They are polite, easy to work with and respond promptly to any concerns. They have a 2-year warranty on their complete 124” engines when supplied with their self-tuning anti-spark knock IST ignition. While expensive, this is great insurance for the engine. S&S parts have a one-year warranty.

However, the customer has an obligation to break in a new engine, avoid lugging the engine as in going too slow for a

particular gear, blowing shifts with no rev limiter and beating up the engine.

Riding under a no-load no-lug situation should always warm up engines since they are air-cooled. The time honored warm up procedure of letting the bike idle is incorrect as there is no cooling air.

Oil changes ensure that removal of oxidized pollutants and debris occurs in a timely fashion and the introducing of fresh engine lifeblood.

Downloads from leading edge modern ignitions like the S&S IST ignition will give an engine rpm history plus oil analysis and damaged parts inspection will tell an accurate story of engine usage. It will also indicate whether there is a parts malfunction or some other situation that is beyond the

owner's control that will lead to a full and comprehensive repair or replacement at no cost.

## The Break-in Procedure

High performance engines, no matter what your philosophy, are intolerant to abuse.

The correct break-in prescribed by S&S will ensure longer engine life and prevent parts from initial damage. S&S Cycle, Inc. recommends the following,

1. On first startup, start up and run the engine for one minute between 1250-1750 rpm.

2. Immediately check for oil return to the oil tank, oil pressure and oil leaks.

3. Do not blip the throttle or place the engine under load as this may damage parts. Head gasket failure is of special concern.

4. Shut off the engine and let cool.

5. Re-start the engine after cooling until the cylinders are warm/hot Fahrenheit), to the touch. Do no leave running for more than 4 minutes. Continue to watch for any problems like oil leaks. Repeat this procedure 4-times. Successive starts should take longer for the cylinders to heat up.

6. Increase the temperature about F each startup.

7. Gently vary the rpm up and down until going up to 2500 RPM on the final startup.

8. Be careful not to let the engine reach full operating temperature.

9. Do not worry about carburetor adjustment because the engine will not be hot enough for accurate settings. S&S presets the carburetor close to final tuning specifications out of the box. IST ignition timing is automatic.

10. I recommend changing the oil and filter at this time. One cannot do this enough during initial break-in.

The first 50 miles is crucial. Rings and pistons are especially sensitive to damage at this time.

11. Do not exceed 2500 rpm while gently varying rpm up

and down to break in the pistons and rings.

I like to ride at night when it is cooler for the engine. Avoid riding in hot weather or in traffic. Do not lug the engine!

12. Change the oil/filter after 50 miles.

13. For the next 500 miles, gradually increase rpms up to 3500 rpm, varying it all the time.

Avoid continuous speeds. Following this procedure is usually annoying to most including myself.

14. Change the oil/filter.

15. The next 500 miles is easier, as rpm's can gradually increase once again. Riding the motorcycle is more normal

but still requires a conservative manner.

No lugging, no drag racing, no Dyno runs, no towing, no sidecars, no loading the engine heavily and no excessive speeds, all of which overheats the engine.

16. Change the oil/filter.

17. After 1000 miles, check carburetor jetting and adjustment.

The engine can now run normally in a sensible manner. Please also see Chapter III: H-D 110 Cubic Inch sections, Engine Tuning, and Break-in.

Conclusion

The mechanic addresses the three bottlenecks to achieving good power in both the Twin Cam 88 and 88B, which are insufficient flowing heads, weak timing chain support system, and emissions controls with the 124-inch build. In fact, every component part coordinates with the other in a complementary and supportive way to produce big power.

Only a few years ago, building a relatively safe engine this big would have been impossible using stock engine casings to support a massive bore combined with a relatively safe stroke to produce 123.6 cubic inches or its metric equivalent of

## S&S Flywheel Identification

The rider can determine the style, stroke, and diameter of S&S flywheels by examining the serial number stamped on the rim of both flywheels in the set. The rider can usually view the flywheel serial number through the timing-plug hole of an assembled engine. Press-together (Twin Cam), flywheel serial numbers are stamped on the inside rim of S&S flywheels. A cylinder removal is necessary to read serial number.

## Interpreting S&S Flywheel Serial Numbers

Sample flywheel serial number

S1234-3 SLX

Big Twin Stroke Code

Big Twin Engine Style Code

* Flywheels machined for special sprocket shaft and stock pinion shaft.

S&S Four Cam (Sportster style) Stroke Codes

* Flywheels machined for special sprocket shaft and stock pinion shaft.

§ Flywheels machined for special S&S pinion and sprocket shafts.

† Flywheel machined to use either 1977 to 1986 style Torrington bearing or 1987 and later style caged roller pinion shaft main bearing assembly.

‡ Machined for use of 1986 and later stock XL connecting rod bearings. 6

I include the codes for all S&S flywheels for rider interest and future reference even though this particular volume is about performancing the Twin Cam.

R&R Cycles PSI Carburetor

Pictured is an R&R Cycles Inc. PSI Genesis Big Air carburetor.

It has a larger float-bowl design to service up to the current biggest V-Twin engine of 155 cubic inches. This unique carburetor has a ribbed interior that intensifies flow characteristics along with vortex generators that control the intake air to align it for maximum port laminar velocity. There are three external adjustments, the pilot, needle, and power jet. Turning these adjusters clockwise will decrease fuel while counter-clockwise turning will increase fuel, all while the bike is running! Superior fuel vaporization keeps fuel in suspension throughout the intake process allows for better throttle response with resultant acceleration. This vaporization process is instrumental in the PSI carburetor meeting 2010 EPA dictates.

## Twin Cam Big Inch Crate Engines

This section is really about JIMS Twin Cam big-inch crate engines since I have already spent much time discussing the S&S unassembled and assembled Twin Cam crate engines.

There are many excellent manufacturers including JIMS (www.jimsusa.com), RevTech (www.customchrome.com), R & R Cycles Inc. (www.rrcycles.com), S&S Cycle Inc. (www.sscycle.com), TP Engineering (www.tpengineering.com), and Ultima (www.jirehcycles.com), among others that manufacture crate engines but most base on the Evolution style engine casings. Very few manufacture crate engines with the Twin Cam style engine casings because of their complexity as well as some proprietary trademark concerns.

JIMS Twin Cam Big Inch Engines

JIMS USA has a 131” Twin Cam engine for touring and Dyna rubbermount models that will bolt into rigid mount Evo style Softail and rigid frames. They do this by changing the

rear motor mount from one that connects to an integral transmission to one that bolts to a traditional center frame down tube appendage. This feature, of course, allows for 1999 and earlier separate transmissions. The reliability of the Twin Cam can now supplant big-inch Evo style engines that while reliable are not as reliable. The stroke remains the same as JIMS 120 inch engine at 4 ½". This is a mere ⅛" larger than the TC96 stroke and a half-inch larger than the stock Twin Cam 88 cubic inch engines.

The bore increases to a massive but very safe 4 or 4.3125".

JIMS: A Safe Stroker

Both JIMS 120 cubic inch engine and its behemoth 131 cubic inch monster use a very conservative 4 ½" stroke so piston speed remains within tolerable limits. The engines achieve their mega-size through big-bore, in Twin Cam engine

casings that can safely handle the bigger spigot holes to accommodate the massive cylinders. The cylinder bores are 4.125” or 4 ⅛” for the 120 cubic inch model and 4 or 4.3125” for the 131 cubic inch engine.

Compression for both is at a very conservative 10:1. These size engines normally increase compression to between 10.25:1 on the low-end and maybe 10.8:1 on the high end for a street bike. With that said, many go higher to over 11:1 compression ratio.

With all other things being equal, which they are not, a higher compression, and longer stroke engine will have more bottom-end torque. What we have here is some low-end power sacrificing for much more reliability.

I suspect that an equivalent S&S engine may be faster off

the line but the JIMS will overtake it in high-midrange rpm and dominate in the higher rpms all things being equal including the size of engine, which again, I must emphasize they are not as so many other factors will adjust my statement up or down.

Thirty years ago, a 96 cubic inch engine was king. Now we have 131 cubic inch engines that are far more reliable than the 96-incher of yesteryear.

CNC ported heads accommodate the increase need for air with 2.080” intake valves.

The JIMS engines use the TC96 oil pump, which increases flow by 10% and scavenging by 23% over TC88 oil pumps. They substitute a JIMS billet cam-support-plate for the stock one. JIMS uses the 2006 Dyna and 2007 later Twin Cam hydraulic cam chain tensioners and timing chains while S&S

replaces these timing chain components by using their proprietary gear drive cams.

JIMS claims a wide powerband using EFI induction. This results with 130 HP and 135 ft-lbs of torque, measuring at the rear wheel. They stipulate quite correctly that results may vary based on induction method whether it is carburetor or fuel injection, camshafts and exhaust. Ratings will also vary on a host of other conditions such as weight–to-power ratio, ignition, ignition timing, and the state of tune of the whole powertrain.

JIMS also makes a 120 cubic inch Twin Cam for either the rubbermount touring and Dyna models or the counterbalanced Softail models. The Softail configuration claims 121 HP and 125 ft-lbs of torque.

The JIMS engine casings are massive, as they will safely support up to a bore of 4.80 inches. The A356-T1 material casings have thicker walls and larger radius corners.

JIMS always assembles the bottom end leaving it to the customer to assemble the rest or to buy the unit as a fully assembled crate engine. Flywheels are trued to .0009”, 10% under .001”, which is very welcome news.

Cam lift for the 120” engine is .653”; the 131 engines uses S.E. 264 cams with a .635” lift. The JIMS I-beam connecting rods are 4340-chromemoly-steel as are the roller rockers and the adjustable pushrods. Chromemoly is serious steel known for its rigid strength and lightweight. The roller rockers rolling pads, the valve springs and the top ball on the pushrods receive full oiling. The ARP 8740 alloy cylinder studs receive heat-treating to 200,000-psi tensile strength. The forged pistons meet NASCAR standards.

Those interested must purchase these engines through a Harley-Davidson dealer. They are for race use only. Nice try… EPA is getting pissed at this disclaimer.

R&R Cycles: 155 Cubic Inch Twin Cam

R&R Cycles Inc. who has an excellent reputation in the H-D aftermarket performance field is building the biggest current Twin Cam engines. They feature tapered Timken bearing sets on both sides of the crank, which is a simple but great and novel innovation that should produce much reliability for crankshaft support.

The engine cases, cylinders, cylinder heads, rocker boxes, and cam covers use wrought billet 6061-T65-aluminum. This material is very strong. It will also produce a gorgeous luster

finish.

Bench flow tested cylinder heads feature hand-blended CNC porting.

The redesigned engine oil passages promise superior oil flow and lubrication.

The gear case cover features a fully baffled crankcase venting system that will prevent most oil carryover. As the reader knows, the bigger you go, the more of a problem that venting oil may become.

The 2-cam 155-cubic inch engine comes with Evolution style motor mounts.

The bore is a whopping 4.600” (4 with a 4.625” (4 ⅝”) stroke, and 10.2:1 compression.

The connecting rod length is long at 8.375 inches and the cams are 620M.

Check out the carburetion, an S&S Super D with three Thunderjets!

The rider has a choice of an EPA compliant or race engine.

An optional R&R PSI carburetor is available.

“The PSI’s new Series 4 production GENESIS “BIG-AIR” carb features include; a much larger billet float-bowl design with more fuel capacity that also offers significant cylinder clearance, an optional dual cable pull, and 3 flat-slide designs

for the ultimate setup on any engine size. This carb works great on motors from 74-100", and rips on motors from 103-117". Monster motors from 124-155" are totally impressed. With the help of R&R Cycles, PSI developed every, unique shape and a ribbed interior that truly intensifies flow characteristics, along with VORTEX Generators that take total control of the air and align it for maximum port velocity… Other features include a shorter throttle pull, a new easier to use external needle adjuster, very tight slide clearance, highly developed slide and needle combinations for incredible throttle response, serious midrange snort, and a smoking Top End! Add that to Rockin' Good LOOKS and a 3-year warranty...Carb adjustments are made by tuning external clicker adjustments…3 external fuel adjustments; Pilot, Needle, and Power Jet…Turn clockwise for less fuel and counter clockwise for more fuel, while the bike is running… No jets to change, all adjustments are made…in seconds…This carb…design (has) superior V-Shaped technology and incredible Volumetric Efficiency! Highly mystified fuel

vaporization keeps fuel in suspension throughout the intake process for incredible throttle response and awesome acceleration! This highly vaporized fuel helps meet 2010 Emission standards thru its 'Clean Burn' process…Accelerate with the response of a 38mm on the bottom, but flows more than a 48mm on top!" 7

It is amazing that aftermarket engine manufacturers can meet 2010 emissions controls with carburetion. I would not think this possible. However, I think that all the engines will lose compliancy once the consumer chooses an exhaust and ignition during installation.

R&R 127 Cubic Inch Engine

This engine has a 4.250" (4 ¼"), bore with a shorter 4.500" (4 ½") stroke. The connecting rod length is 7.690" with a higher compression ratio of 10.8:1 and 620T

camshafts. Carburetion is with an S&S Super G with a single Thunderjet, which is a traditional combination.

## R&R 139 Cubic Inch Engine

This engine has a 4.375” (4 ⅜”) bore with a 4.625” (4 ⅝”) stroke. The connecting rod length is 8.000” with a compression ratio of 10.5:1 and 620M camshafts. Carburetion is with an S&S Super D with three Thunderjets.

## R&R 147 Cubic Inch Engine

This engine has a 4.500” (4 ½”), bore with a 4.625” (4 ⅝”), stroke. The connecting rod length is 8.00” with a higher compression ratio of 10.5:1 and 620M camshafts. Carburetion is with an S&S Super D with three Thunderjets.

## Conclusion

The Twin Cam design allows big-inch engines to progress into area that most of us never thought possible.

Not only can the stock Harley-Davidson Twin Cam engine cases easily support 124 cubic inches safely but also aftermarket ones that are wider to accommodate even larger spigot holes for bigger bore cylinders currently up to 4.80”. As this type of case manufacturing progresses, do not be surprised if Twin Cam engines broach 170 cubic inches and beyond.

The constraint of a 4 ⅝” stroke to keep potentially damaging piston speeds in check limits this way to safe bigger displacements but not so, with big-bore where piston speed remains the same.

1 S&S Cycle, Inc. 2007. Performance Guide For 1999-06 Big Twins with Twin Cam 88® and Twin Cam 88B™ S&S Cycle, Inc. catalogue. Page 16.

2 Ibid., Page 17.

3 Doherty, Tim. Doherty Machine. Part of interview with Tim Doherty 2006.

4 Ibid.

5 Ibid.

6 S&S Cycle, Inc. Special Instructions for Fully Assembled S&S® Flywheel Sets

EXCEPT FOR S&S® PRESS-TOGETHER FLYWHEELS.

Instruction 51-1019. Page 3-4.

7 R&R Cycles Inc. Website, (www.rrcycles.com). Page,

Performance Accessories, sub-page, RR / P.S.I. Carburetors.

# Chapter XIII

## Forced Induction

Forced induction is the opposite of natural aspiration.

### Natural Aspiration

A naturally aspirated engine is one that relies on the laws of physics to feed the combustion chamber with air. I am not sure why how the word aspiration became to mean the induction of air since the dictionary refers to ambition and goals. In any case, aspiration for our purposes defines as the delivery of air, or more properly oxygen into the engine.

Carbureted engines rely on vacuum and other physical laws to draw in air as well as the fuel.

EFI engines take control of fuel delivery and feed the fuel under pressure in precise incremental amounts into the combustion chamber but still rely on natural aspiration for air delivery.

## Forced Induction

Forced induction will send greater quantities of air and thus oxygen than the laws of physics can deliver naturally into the combustion chamber.

Supercharging forces air into the engine via centrifugal force, an impeller, or screws driven by the engine itself. An impeller is a precision enclosed propeller that boosts the

volume of air.

Turbocharging forces air into the combustion chamber, again by an impeller driven by the engine's exhaust gases.

Nitrous oxide delivers into the combustion chamber under pressure from a pressurized external bottle.

Supercharging, turbocharging and nitrous oxide do not replace carburetion or electronic fuel injection but act in conjunction with either.

Ram Flow

Ram flow manifolds and air cleaners that face into the wind also stack greater amounts of air into the chamber that

will allow more gas to burn producing more power. However, the process still involves a natural induction by taking advantage of rushing air past a moving motorcycle. For more information on ram flow and naturally aspirated air delivery, see Chapter IX: Fuel sections, Air Filtration and Power, and Velocity Stacks, and Performance Filters and Air Breathers, and Ram Flow Induction Systems.

## Air is King

The old adages Go big or go home and Nothing beats cubic inches are a few maxims that express the opinions of those that feel bigger displacement engines is the only way to go for an increase in performance. The Go Big axioms are true in the sense that more cubic inches will obviously increase power. However, cubic inch increases do it through quantity that sometimes lacks the quality of getting every bit of oomph possible per cubic inch.

What is not true is that it is the only way. Nor is it true that nothing can beat cubic inches. Furthermore, all the ways of making an engine faster do revolve exclusively around getting more air into the engine.

This begs the question, why more air. The reason is for more gas burning capability, which creates heat that expands creating pressure to drive the piston down thereby turning the crankshaft and away we go.

The reader must get sick of me writing the next two sentences but what can I do? Heat and pressure is the same thing. More gas will not burn without more air. Therefore, it is not enough to pump more gas into the engine because an exact amount of oxygen is required to burn a given amount of fuel. These chemically correct mixtures revolving around

about a rich 12 parts air to a lean 15 parts air to one part fuel depending on type of riding and emissions controls is essential for an engine to operate efficiently. The bottom line is that to burn more fuel; there will be a requirement for more air. There are three main ways to increase air delivery.

1. The first is to make the existing engine-breathing tract more efficient. This involves higher lift, longer duration and longer overlap cams as well as larger valves coordinating with porting and polishing of the heads and modifications to the fuel supplier, whether it is carburetion or EFI. An exhaust system is capable of adding or detracting power via its breathing capabilities.

2. The second way of course, is increasing cubic inches.

3. The third way is via forced induction: supercharging, turbo charging or nitrous oxide.

Of course, in real life, proficient engine builders do not separate the above techniques but may combine some or all them to create the fast machines.

I will now discuss another facet of air volume and that is volumetric efficiency.

## Volumetric Efficiency

Instead of enhancing swept volume to increase power like I discussed in the last chapter, Chapter XII: Cubic Inches: Stroker and Big Bore, I am going to look at another separate but related methodology called volumetric efficiency. Volumetric efficiency refers to the efficiency with which the engine can move the air/fuel charge into and out of the cylinders. More correctly, volumetric efficiency is simply a

ratio usually expressing as a percentage of the volume of fuel and air that actually enters the cylinder during induction to the actual capacity of the cylinder under static conditions.

Static means at rest without the benefit of induction enhancements like the high rpm scavenging forces of momentum. Therefore, if an engine can create above ambient induction manifold pressures, the engine will derive a volumetric efficiency above 100% cylinder fill. Volumetric efficiencies can improve in a number of ways such as increasing valve sizing relative to the volume of the cylinder and through porting the intake port and porting and polishing the exhaust port. Engines with higher volumetric efficiency will generally be able to run at higher rpm and produce more overall power. I discuss many times that it is possible to achieve up to 130% cylinder fill or volumetric efficiency via internal scavenging principles.

However, I will look at increasing volumetric efficiency from external sources by forcing, compressing, ramming, or stacking more air into the cylinders. I can do this easily with a few nifty additions known as supercharging or turbocharging or nitrous oxide.

## Wipe the Knowledge Slate Clean with Forced Induction

Forget all you know about normally aspirated engines in terms of cams, cam timing, cam overlap, duration air delivery, mechanical lag time, ignition lag time, and a bunch of other stuff.

Forced induction results in pressurized air entering the combustion chamber. The air cannot escape out through an open exhaust valve or reverse back through an open intake valve without losses in power and efficiency. Cam overlap

keeps the exhaust valve open during the initial stages of the intake cycle and the intake valve opening during the previous exhaust cycle. This is necessary to allow for the mechanical lag time it takes parts to start moving plus, as in the case of valves, time for them to open sufficiently to permit the passage of incoming or outgoing gases. Overlap also allows scavenging to occur that purges the combustion chamber of spent gases and draw in the fresh intake charge.

Forced induction needs to curtail valve overlap and cam duration but increase cam lift and thus valve lift for power generation and running efficiencies. This is very different from what natural aspiration requires.

In short, the four strokes necessary to complete one engine cycle, valve opening and closing times need to reflect theoretical cycles for forced induction while the durations of

natural aspiration may go up to 260-degrees or more of overlapping engine cycles.

Forced induction is all about containing the compressed intake charge in the cylinder because its pressure is greater than ambient atmospheric pressure. Forced induction cams will open and close valves closely in tandem with 180-degree cycle beginnings and endings.

## Supercharging

A supercharger is an induction device that pressurizes the air intake charge above atmospheric pressure. Turbocharging and supercharging are very different delivery systems that utilize the same basic technique to achieve much more concentrated flows of air than natural aspiration.

Air at atmospheric pressure will suck into the combustion

chamber during the intake stroke of a four-cycle engine. Therefore, there is a limit to the amount of air ingested during the natural aspiration of induction.

Of course, fuel requires air in certain specific proportions to burn. There is no point in attempting to burn more fuel unless the air intake increases. Supercharging fulfills this function admirably.

Turbochargers and superchargers increase intake by compressing air above atmospheric pressure, without the benefit of the physics principle of vacuum. This forces more air into the engine, providing a boost. With the additional air in the boost, more fuel can add to the charge with a resultant increase in power. Both superchargers and turbochargers accomplish this. In fact, the term turbocharger is a shortened version of its original name that has separated into two

distinct meanings over time. Both act as an air compressor, which may be an enclosed fan (impeller), meshing rotors or interfaced rotating screws that increase air pressure.

With the additional air in the boost, more fuel can add to the mixture thereby increasing the power of the engine.

## High Altitude Supercharging

In high-altitude situations, where engine performance deteriorates because the air has low density and pressure, turbo/superchargers will deliver higher-pressure air to the engine so it can operate optimally. This is why superchargers are common on many internal combustion airplanes, which makes sense when you consider that airplanes spend most of their time at high altitudes, where significantly less oxygen is available for combustion.

# Parasitic Loss of Horsepower

Superchargers use up some mechanical engine energy also known as parasitic loss of horsepower because the engine must also drive the supercharger via a belt/pulley system attaching to the engine output or sprocket shaft on the left side of the engine primary system. Unlike turbochargers, which use the combustion exhaust gases to power the compressor, superchargers draw their power directly from the crankshaft.

Most drive by an accessory belt, which wraps around a pulley that connects to a drive gear. The drive gear, in turn, rotates the compressor gear. The rotor or impeller of the compressor can come in various designs. However, its job is to draw air in, compact or stack the air into a smaller space and discharge it into the intake manifold.

Superchargers can spin up to speeds as high as 65,000 rpms, which is slower than a turbo but still very fast. Superchargers have a turbine drive gear that turns a smaller air pump compressor gear. The high rpms achieve by gear reduction in designing the turbine drive gear larger than the air pump compressor gear, which causes the compressor to spin faster.

Both turbo and superchargers achieve performance gains the same way by increasing the density of the air/fuel charge beyond atmospheric pressure within the combustion chambers of an engine.

Atmospheric Pressure

Atmospheric pressure is the weight of the air at sea level.

All the air above creates a downward pressure because of gravity. This gravitational weight measures in pounds per square inch (psi). One atmosphere is the atmospheric pressure of the air surrounding the earth at sea level, which has a theoretical pressure or weight of 14.69 psi. This figure often rounds off to 15 psi, with which, the reader may be more familiar. I use the term theoretical because gravity is greater in some areas than others and finding sea level is not as easy as one might think. Factors like the moon's gravitational pull, temperature changes, and the ever-changing tides complicate this.

## The Weight of Air and its Relationship with Power

Colder air weighs more than warm air. This increase in density occurs because the air molecules pack in closer when cold. Air molecules also expand with heat and become less dense.

An air charge refers to the air going into an engine. Therefore, the temperature of the air charge is also important from an efficiency breathing perspective as well as a performance one, since a cooler charge is denser and therefore more powerful. Density achieves in two ways, via a cooler temperature and the forced induction of more air in than the normal atmospheric weight. Turbochargers and superchargers both increase air density by forcing additional amounts of air beyond atmospheric pressure at the lowest temperature possible.

Turbo and Supercharging Reliability and Efficiency

Properly set up, turbochargers and superchargers can produce excellent performance using low octane gasoline. Reliability is excellent. Fuel economy remains good with

normal riding. Furthermore, emissions controls are still relatively easy to meet with these units. What else can the rider want?

In the old days, Harley people found it necessary to adapt superchargers and turbochargers from cars and other machines. As a result, reliability suffered. The adapting units were often from different sized engines. This lone factor results a myriad of problems. The next difficulty was installing the charger and it's plumbing onto an engine for which they were not designed. This becomes an expensive machine shop nightmare. If done incorrectly and/or with shortcuts, breakdowns and even engine damage could well ensue.

The present day situation is much different. Now, most Harley-Davidson's have design fitment for these air density amplifiers. For the most part, they are easy to install. Maintenance is simple.

Forced induction methodology is very much simpler relative to internally performancing a naturally aspirated engine. This can also be relatively inexpensive to developing the same horsepower and torque, as traditional methods will.

## Boost and Power Expectations

The turbo/supercharged engine produces much more power overall than the same engine without the charging. A typical boost provided by either a turbocharger or a supercharger is 6 to 8-pounds per square inch. Since normal atmospheric pressure is 14.7 psi at sea level, it follows that the engine cylinders are getting approximately 50-percent more air packed in. Therefore, one might expect to get 50% more power. However, like most mechanical apparatus, neither type of charger is perfectly efficient so a 30% to 40%

improvement is more realistic.

The key difference between the two chargers is the power source. Exiting exhaust gases from the engine acts as the power supply to run the turbocharger air compressor. More correctly, the exhaust runs through a turbine, which in turn spins the compressor. The supercharger mechanically drives via a pulley and belt system off the Twin Cam's power sprocket-shaft located inside the front of the primary cover.

However, there are tradeoffs in both systems. In theory, a turbocharger is more efficient because it is using the wasted energy in the exhaust stream for its power source. On the other hand, a turbocharger causes some amount of backpressure in the exhaust system. This tends to result less boost until the engine is running at higher rpms.

The rotor of the compressor can come in various designs,

but its job is to draw air in, squeeze the air into a smaller space and release the pressurized intake charge into the intake manifold.

Some manufacturers claim specific horsepower/torque increases for customer ease of understanding but turbo/ superchargers actually add horsepower as a percentage gain. Percentage gain relates to the percentage of an atmosphere increase over ambient atmosphere or boost. Of course, deducting an inefficiency factor will be more accurate as some of the boost will invariably escape the combustion chamber through a valve. Theoretically, if a turbo/supercharger induces another atmosphere or 14.7 psi of boost, an engine with stock compression and a stock emissions cam, engine output will essentially double, all other factors being equal. In reality, after adjusting for thermal and mechanical energy losses, an efficient supercharger generating 7.5 psi of pressure or boost will gain around 35% to 40% horsepower and torque

over its naturally aspirated equivalent.

Mechanical losses result from the engine powering the supercharger. Thermal losses come from the compressed air heating up and therefore becoming less dense.

There are three types of superchargers, Centrifugal, Roots, and Twin-screw. The first original superchargers were of the Roots type. The main difference between the various types is how the air pressurizes and delivers into the intake manifold of the engine. A Centrifugal Supercharger uses an impeller to draw air in while the Roots and Twin-screw Superchargers use different kinds of meshing lobes. Although all three designs provide a boost, they differ considerably in their efficiency.

The type of riding will dictate the size supercharger necessary whether it is for street or racing.

# Roots Supercharger

The Roots supercharger is the oldest and designed by the Roots family circa 1858. The invention of cars or motorcycles was still decades away.

The first superchargers, using the Roots Supercharger design was for the ventilation of mineshafts. Its first use on a car was by a company called Daimler, which of course is currently recognizable as Daimler Chrysler, in Germany in the year 1900.

As meshing internal lobes spin, using much the same principles as the Twin Cam oil pump, air traps in the pockets between the lobes that progressively narrow, compressing the air. The lobes carry the compressing air from between the fill side to the output side. Large quantities of air deliver into the

intake manifold, stacking up to create inductive pressure.

Origin of the Blower

Simplistically, Roots Superchargers are basic air blowers. Hence, the term blower was born and is now widely used to describe all three types of superchargers. However, the Roots system is the least efficient supercharger for three reasons.

1. First, it adds much more weight to the vehicle. Muscle car people can be similar to Harley riders in that the large bulk of a Roots or Twin-screw Supercharger sitting on top of the engine, sticking through the hood looks cool to the genre.

2. The second disadvantage to the Roots Supercharger moves air in bursts instead of a regulated and continuous flow.

3. Furthermore, Roots Superchargers will decrease fuel economy because of their drive power requirements.

However, they generate more power in the lower rpm range, which may be an advantage or a disadvantage depending on intended use.

## Twin-screw Supercharger

A set of intertwining lobes that resemble worm gears pulls air through the Twin-screw Supercharger. The meshing lobes conically taper from the intake end to the discharge. Once again, air traps between the pair of tapering rotor lobes, progressively compressing the air from the intake side to the discharge end.

The Roots method pressurizes the air into the manifold

while the Twin–screw compresses the air inside the rotor housing. This is the mechanical advantage that the Twin-screw Superchargers have over the Roots design.

Their immediate disadvantage is that they cost more because of manufacturing precision of the conically tapering lobes or screws. Another is that this type of supercharger creates an annoying to some, whine as the air discharges from the outtake side of screws. This will require a noise suppressor if an irritation. However, for the Harley noise-lovers in the crowd this soon becomes an advantage with the bulk in a world where fat is beautiful.

The Twin-screw Supercharger, like the Roots develops more power in the low rpm range.

Centrifugal Supercharger

The third and most efficient supercharger of the three is the Centrifugal Supercharger that powers an impeller, which best describes as a sophisticated and precise fan. The impeller rotates at very high speeds up to 65,000 rpms. This level of rpm, unfamiliar to the Harley world, quickly draws air into small compressor housing. Centrifugal force generating from the high-speed rotation of the impeller, takes air drawn into its center forcing it outwards.

The high-speed outward radiating air leaves the impeller with an opposing low pressure. This corrects via a set of fixed vanes surrounding the impeller. The vanes slow down the speeding air. This slowing force converts the former low pressure to turn into high pressure.

Centrifugal superchargers are the most popular of the three types because of their efficiency and size. They are currently

and for good reason the most common of all forced induction systems in the automotive world. They are smaller but not small, lighter but not lightweight and attach to the left side of the Harley engine where it is close to its power source, the crankshaft as it protrudes into the primary system. Centrifugal Superchargers also create a distinctive whine as the engine revs up forewarning other H-D riders that this is a bike to reckon with.

A big bonus is that the Centrifugal Supercharger produces big increases in power using much the same amount of gas as does the naturally aspirated engines, without hard throttle cranking. However, as always, when the throttle cranks, fuel economy will decrease. This is obvious since any performanced engine including those that are turbo-supercharged have the ability to consume additional gas to produce additional horsepower.

It is just that the turbo-supercharger exponentially increases power. The Centrifugal Supercharger is excellent in high rpm situations because this is where it generates much of its power.

## Supercharger Benefits

### Power

The obvious first benefit is the dramatic increase in power. Adding a supercharger, essentially delivers the power of a larger engine with a smaller one.

## No Lag Time: Immediate Throttle Response

A supercharger does not suffer the lag time common with many turbochargers. Lag time describes the delay encountered when twisting the throttle and the response in developing power. A turbocharger lag time results because higher rpm needs to generate higher velocity exhaust gases to drive the impeller thus there is a delay after throttle turning. In fairness, modern turbochargers attempt to deal with the lag time issue, many times successfully. However, the supercharger does not require extra engineering to achieve immediate throttle response.

## Low or High End Power

As I have said, the Roots and Twin-screw Superchargers develop more power at lower rpms while the Centrifugal Supercharger is most efficient at developing power during higher rpms. The reason for this is that the centrifugal impeller produces boost with higher speeds while the other two favor low-end grunt work.

## Easy Installation

On a car, the supercharger is easier to install and requires less plumbing like exhaust system modifications. Modern Harley-design superchargers and turbochargers are both simple to mount and include at least the exhaust headers to facilitate installation ease. In other words, both are available in kits for specific models that include all mounting hardware,

necessary support parts, and the exhaust

While there is not a wide variety of choice, there are different manufacturers from which to choose. I will example one for each system of forced induction as I proceed. They are only examples and not necessarily the best for the reader's application and needs.

External Oiling Source

Superchargers do not require an external oiling source. Therefore, shutdown of a supercharger is as simple as turning off the key and thus ignition. Turbochargers generally require external oiling. Therefore, they should idle for about thirty seconds before shutdown to allow lubricating oil to cool. This is a generalization; always consult the provided instructions with the turbocharger in use.

## Warm-up

Superchargers require a warm-up before cranking the throttle…and what engine does not? However, they function best with normal operating temperatures.

## High Altitude Functionability

Superchargers are a huge success in operating internal combustion engine airplanes at higher altitudes. There is less oxygen at altitude but the supercharger takes what is available and compresses it to develop engine performance not available with normal aspiration.

## Disadvantages of Supercharging

The biggest disadvantage of superchargers is also their defining attribute.

## Parasitic Loss of Power

I can say the same of turbochargers whose distinctive characteristic is using formerly wasted exhaust gases as its power source. The engine crankshaft drives the supercharger. Therefore, the supercharger must steal some of the engine's torque. A supercharger can consume as much as 20-percent of an engine's total power output. However, a supercharger can generate 45-percent and beyond additional horsepower. Obviously forced induction proponents feel that the trade-off is a positive one and well worth it.

## Engine Strain

Supercharging will add strain to the engine. The Twin Cam is of such superb construction that this is not an issue in my mind but it might be of concern in some predecessor Big Twins like the Shovelhead. The combustion chamber, pistons, connecting rods, and bottom end components like bearings and crank pin bear-the-brunt of boost oxygenation and the resultant increase in heat and thus pressure. Older Harley-Davidson's will need upgrading of the above parts for sustained supercharger and turbocharger usage.

Expense

I suppose another disadvantage of forced induction is the upgrading expense. The use of higher-octane gasoline is another expense that normally aspirated engines also require.

Despite their disadvantages, turbo-superchargers are still a very cost-effective way to increase horsepower.

# Detonation

The ugly face of detonation is not only a factor with every modern Harley and thus needs controlling when utilizing a turbo or supercharger also. The great engine destroyer, detonation, is a condition with which we are all familiar. The sound of pinging or knocking that can best describe as ball bearings rattling round inside the engine cylinders usually occurs under engine load or run-on when the ignition first turns off.

When detonation occurs, the internal pneumatic forces can accentuate ten times the normal forces acting upon a properly operating high performance engine. Excess heat may result inside the cylinder from compression, low octane gas: lean air-to-gas ratio and/or advance spark timing. Detonation is fuel

pre-ignition. Excess heat in the combustion chamber may cause a spontaneous combustion at the wrong time and usually the wrong place.

Detonation occurs when the piston is on its upward journey on its compression stroke or during the following power stroke. Detonation is a powerful heat creating pressure that sometimes is trying to expand in a cylinder chamber that is shrinking in size due to the rising piston. In short, this force sometimes attempts to reverse the direction of the piston and the engine. Of course, detonation cannot do this but it can wreak havoc on the structure of the piston dome and upper piston ring lands.

As an engine generates more power through performance modifications, additional heat produces. Heat is pressure and pressure drives the piston down to create power.

## Forced Induction Produces Less Heat

Although forced induction engines usually produce far less heat than comparable naturally aspirated high compression engines, detonation can still be a problem.

## Controlling Detonation

On a naturally aspirated motor, increasing the gas-to-air ratio will provide a cooling effect as will retarding the ignition timing, lowering compression and/or raise the gasoline octane rating. Retarding ignition timing and lowering compression will also reduce engine power as will a too-rich AFR. I discuss detonation extensively in Chapter III: H-D 110 Cubic Inch sections, EPA Perfection: Stoichiometric Air Fuel and Solving the Tendonitis of Lean Burning Also, in Chapter VI: Oil sections, Detonation: The Great Engine and Heat is the

Enemy, and Lugging and Detonation. In addition, please read Chapter VII: Torque and section, Compression as well as in other places.

Pump gas will only tolerate a limited amount of heat before pre-ignition occurs and causes detonation. Although forced induction engines usually produce less heat than comparable naturally aspirated higher compression engines, the cylinder temperatures in intercooled engines are very much cooler again.

## Intercoolers

As the air compresses, it generates heat because pressure and heat are synonymous.

### Hotter Air Loses Density

The expansion of hotter air loses density. Therefore, less-densely oxygenated air cannot expand as much, thereby reducing potential power gains. As higher boost levels increase, the heat generation of compressing air will produce diminishing returns. As boost increases, so does pressure, which means increasing heat will expand the air molecules losing progressively more density.

## Racing Fuel

Racing fuel for street riding is one unwieldy option since it is not readily available and is a hassle all the way round.

## (Inter) Cooling the Hotter Air

The performance mechanic will recognize this heat expansion problem and take steps to rectify it by cooling the

compressed air after it leaves the discharge side of the turbo-supercharger before it enters the intake manifold. Intercooling aptly describes this process.

There are two different types of intercoolers,

1. Air-to-air.

2. Air-to-water.

Intercoolers work much like a radiator. Cooler air or water depending on the type of intercooler flows through tubing. The exiting turbo-supercharger hot air flows through and around the cooler tubing thus lowering the hotter air temperature increasing the air density before it exits the intercooler and enters the intake manifold.

Therefore, a denser, highly oxygenated compressed air

enters the combustion chamber to do its thing. Intercooling is primarily an efficiency prerequisite. It is a necessity for running higher boost levels while maintaining reliability.

## 100% Increase in Power

Although I think a street bike and rider should be content with a 40% to 45% increase in power from turbo-supercharging it is possible to achieve a 100% increase with around a 17 psi boost on a Twin Cam engine with stock 8.8:1 to 9.2:1 compression running pump gas without the benefit of racing fuel. I am not sure why I said that a rider should be content with less than what is possible except that 17-psi boost is an awful lot.

Intercooling does double-duty by actually removing heat that causes detonation.

Higher boost levels produce much more heat and are not possible without intercooling. Furthermore, an intercooler will safely allow use of higher boost levels in conjunction with fully advanced ignition timing on readily available pump gas.

## Intercooling Produces More Power

Because intercooling removes heat, increases air density allowing more fuel for burning, and permits the use of Factory or near-Factory ignition timing, an intercooled turbo/ supercharger system will produce larger power gains than a non-intercooled system, especially for fuel-injected motors running street-pump gas. The bottom line is that intercooling allows maximum performance with maximum engine component protection.

## Fuel Requirements

The issues that determine the turbo-supercharger fuel requirements are type of induction system, injected or carbureted, engine compression ratio, and whether intercooled or not. Lower compression motors can run more boost and therefore compress more air to produce more power. This is one time it is better not to increase the Twin Cam's compression ratio as this necessitates the usage of lower boost levels.

## Carburetors Handle More Boost than EFI

Carbureted engines can handle more boost than EFI equipped Twin Cams because of the cooling nature of liquid gas. Carburetors deliver most of the gas in a liquid state, and as this raw fuel atomizes from liquid to gas, an endothermic reaction occurs, which draws heat and cools the incoming air.

Endothermic means to absorb heat. This is why a carbureted motor can safely handle more boost than a comparable fuel injected motor.

As I have said, pump gas is fine with either superchargers or turbochargers but try to put in the highest-octane gasoline available.

## Cams for Superchargers

Forced induction does not like cam overlap.

Overlap is the degrees crankshaft rotation that the exhaust valve stays open after TDC during the same time in degrees of crankshaft rotation that the intake valve opens BTDC (before top dead center) on the exhaust stroke. I explain overlap at length in Chapter IV: Cams, Cams, Cams, sections, Valve and Cam Lobe Centerline Angle (LCA), and Cam Lobe Separation

Angle (LSA).

Less Cam Duration and Overlap

A supercharged engine with too much overlap will cause reversion of the charge to either exit the exhaust or go back through the intake tract and out the air breather. Thus, overlap will decrease cylinder fill on a supercharged engine because the charge will push out the exhaust tract through an open exhaust valve that needs closure or push back through the intake port through an open intake valve that also needs earlier closure.

Thus, the exhaust valve needs to close sooner on its cycle and not extend opening into the intake cycle. The intake valve should not open too far before its theoretical cycle begins. In other words, the intake valve should not begin opening too

early in the closing degrees of the previous exhaust cycle.

## Forced Induction and Forced Leakage

Overlap works wonderfully in a naturally aspirated engine as the physics Principle of Momentum scavenges out exhaust gases while pulling in the fresh intake charge, particularly at higher rpms. Forced induction pushes in the compressed charge so forcefully that open valves will cause forced

## Low Compression

Supercharged engines also do not like high static compression ratios.

The Twin Charger (supercharged and turbocharged), Golf automobile pulls off a relatively high ratio but there are more cylinders and a very sophisticated ECU computerized engine

management system to compensate.

The problem with a higher static compression ratio is that it may combine with boost to increase the effective compression ratio too much causing the great engine destroyer detonation.

## Stock Emissions Cams Work Well with Supercharging

Stock cams have emissions features with little overlap. Therefore, a stock cam works well with a supercharger. Twin Cam stock camshafts have relatively high lifts and low duration.

## Performance Supercharger Cams

The respected cam manufacturer, Leineweber (www.leinewebercams.com), supplies high lift, low overlap cams for supercharging purposes. Jim Leineweber has more of a selection for Evolution and predecessor model Harley-Davidson's'.

For a Twin Cam, I recommend their T3S camshaft with .510" valve lift, intake timing of and exhaust timing of producing a small overlap of Duration is on the intake and degrees on the exhaust with cam lobe centerlines of intake and exhaust producing a lobe separation angle of Stock valve springs are okay with these camshafts but as I say in the next section the intake works better with heavier springs. The T3S is available for chain drive or gear drive.

I would choose a gear drive system especially for 1999 to 2006 Twin Cams.

Forced induction engines, which include supercharging, turbocharging and nitrous oxide, require heavier seat pressure valve springs on the intake valves. In fact, seat-pressure requirements increase in tandem as boost increases. Cam life reduces with heavier valve springs. Hydraulic roller tappets like those found in Harley-Davidson's cope well with the increases in spring pressure relative to automotive flat tappets.

Exhaust valves have a different problem in forced induction engines. The increase in cylinder pressures attempt to hold the exhaust valve closed and resistant to opening when the cam lobe dictates. This puts pressure on the cam lobes initiating faster wear characteristics. Again, the Harley-Davidson type hydraulic roller lifters help immensely in

assuaging these wear concerns. Engines that use flat tappet cam lobes need to change to roller cams and roller tappets. Please refer to a later section in this chapter, Turbocharger Cams for more information.

## Intercooled Centrifugal Supercharger

Accessible Technologies, Inc. (ATI) manufactures the integrated Intercooled Centrifugal ProCharger Supercharger for Twin Cams. The ProCharger design is for either stock or modified Twin Cams. It is an excellent and well thought out gear-driven supercharger with self-contained oiling that does not affect motorcycle drivability, rider position, or idle quality. Installation time is between four to six hours for a competent mechanic. ATI has a limited one-year warranty and an available 3-year warranty for an extra $49.00 at the time of writing in June 2008.

The ProCharger promises a bolt-on kit that will provide 50% to 65% gains with stock or modified engines. ATI has intercooled systems for TC88 and TC96 applications and are 100% complete, including injectors and tuning module. Their intercooled Tuner Kits are for modified and larger displacement applications and injectors and tuning module. A polished finish is standard with an available black finish. A front-mount intercooler is standard and there is a choice of four air filter covers, and two supercharger air inlets at no extra cost.

As the ATI website explains, “All superchargers operate with an intake manifold temperature in the general range of above ambient (outside air) temperature at 8 psi. At the same boost level, an Intercooled ProCharger operates at only above ambient...no supercharger alone will ever begin to match the system efficiency of an intercooled supercharger system. This is simply because compressing air creates heat, as dictated by

the laws of physics (Boyle's Gas Law). Even in the case of "perfect compression" (100% adiabatic efficiency, which is physically impossible without an intercooler), air temperature would increase by approximately at only 8 psi, while the lower (40-80%) efficiencies of all non-intercooled superchargers produce substantially higher temperatures." 1 Adiabatic defines as changes in temperature caused by the expansion or compression of a body of air as it rises or descends in the atmosphere, with no exchange of heat with the surrounding air.

ATI continues, "The amount of heat produced (adiabatic efficiency) by each supercharging technology is what determines the boost limitation...Intercooled ProCharger systems are the fundamental, OEM solution - because not only is less heat created when the air is compressed, the majority of this heat is actually removed through intercooling. The bottom line is that intercooled boost is substantially more

powerful and safe for your engine than hot, non-intercooled boost. Now that technology has developed to the point that gear-driven superchargers are powerful enough to blow reliably through an intercooler, it simply does not make sense not to intercool, especially for fuel-injected applications running pump gas. In fact, for high compression engines or continuous duty applications, such as marine or towing, intercooling is essential for reliability.

- Full Timing: this reduction in temperature allows you to run Factory (or close to Factory) timing, and avoids the substantial horsepower loss inflicted by excessively retarded ignition timing

- More Low-End Boost and Horsepower: the intercooler also acts as a passive wastegate, flattening the boost curve at higher rpm's and allowing more boost to be dialed in at lower rpm's

- An Expanded Power Band: full timing and forced induction keep the engine pulling hard to the redline

- More Boost: not only will you experience the above benefits at any boost level, you can also safely run substantially more boost when intercooled!" 2

Clearly, the only type of performance that matters is reliable performance, and detonation is the single biggest threat to engine reliability. The boost range for reliable performance, without detonation, can determine by looking at and considering the type of supercharger technology, and the compression ratio of the motor. With a lower compression ratio, an engine can safely handle more boost, everything else being equal. Similarly, if the temperature of the compressed air is lower, an engine will have a much higher detonation

threshold...and will be able to safely handle more boost. 3

## Twin Cam Touring EFI ProCharger Kits

Harley-Davidson Twin Cam touring FLT model complete kits are available for 2007 with intercooler for EFI; 2002 to 2006 with intercooler for EFI and 1999-2001 EFI intercooler kits. Intercooler Tuner Kits are available for the same year EFI models for modified Twin Cam engines without injectors and the tuner module. All kits are side mount with front mount intercoolers and promise 50% to 65% horsepower gains.

## Twin Cam Softail EFI ProCharger Kits

Harley-Davidson Twin Cam Softail model complete kits are available for 2007 to present with intercooler for EFI and 2001 to 2006 with intercooler for EFI models. Intercooler

Tuner Kits are available for the same year EFI models for modified Twin Cam engines without injectors and the tuner module. All kits are side mount with front or side mount intercoolers and promise 50% to 65% horsepower gains.

## Twin Cam Dyna EFI ProCharger Kits

Harley-Davidson Twin Cam Dyna model complete kits are available for 2006 to present with intercooler for EFI and 2004 to 2005 with intercooler for EFI models. Intercooler Tuner Kits are available for the same year EFI models for modified Twin Cam engines without injectors and the tuner module. All kits are side mount with front or side mount intercoolers and promise 50% to 65% horsepower gains.

## Twin Cam Touring Carbureted ProCharger Kits

Harley-Davidson Twin Cam touring FLT model complete

kits are available for 1999 to 2006 with intercooler for carburetor. Intercooler Tuner Kits are available for the same year carbureted models for modified Twin Cam engines without injectors and the tuner module. All kits are side mount with side mount intercoolers and promise 50% to 65% horsepower gains.

Twin Cam Softail Carbureted ProCharger Kits

Harley-Davidson Twin Cam Softail model complete kits are available for 1984 to 2006 with intercooler for carburetor. Intercooler Tuner Kits are available for the same year carbureted models for modified Twin Cam engines without injectors and the tuner module. All kits are side mount with front or side mount intercoolers and promise 50% to 65% horsepower gains.

Twin Cam Dyna Carbureted ProCharger Kits

Harley-Davidson Twin Cam Dyna model complete kits are available for 1999 to 2005 with intercooler for carburetor. Intercooler Tuner Kits are available for the same year carbureted models for modified Twin Cam engines without injectors and the tuner module. All kits are side mount with front or side mount intercoolers and promise 50% to 65% horsepower gains.

## Turbocharging

As I said earlier, the term turbocharger is a shortened version of its proper and original name. I have already had many discussions in this chapter on turbocharging and will not repeat them here In order to achieve the boost of compressed air, the turbocharger uses the exhaust flow from the engine to spin a turbine, which in turn spins an air pump.

The turbine in the turbocharger spins at speeds of up to 150,000 rotations per minute (rpm), that is over 25-times faster than Twin Cam or XL engine rpm capabilities. Since a turbocharger hooks up to the exhaust, the temperatures in the turbine are very high.

Inefficiency comes from the fact that the power to spin the turbine is not free. Having a turbine in the exhaust flow increases the restriction in the exhaust. This means that on the exhaust stroke, the engine has to push against a higher backpressure. This subtracts a little power. However, this loss is not nearly as much parasitic power loss that occurs with the crankshaft-powered superchargers.

Trask Turbocharger

The pictured Trask Performance Softail turbocharger will work on 2000 and later Softails and 1999 and later Dynas. Carbureted or injected, this kit is easy to install and makes great power with 115 HP and 120 ft-lbs torque on a stock engine with 8 lbs of boost on 91-octane pump gas. Trask also offer an optional intercooler kit. Each kit supplies with a highly modified Garrett GT Series turbocharger, billet plenum with built-in velocity stack, billet breathers, billet cam cover, ceramic or chrome pipes, hi-flow injectors and a performance Barnett clutch spring. I highly recommend an intercooler. As the air compresses, it generates heat because pressure and heat are synonymous. The expansion of hotter air loses density. Therefore, less-densely oxygenated air cannot expand as much, thereby reducing potential power gains. As higher boost levels increase, the heat generation of compressing air will produce diminishing returns. As boost increases, so does pressure, which means increasing heat will expand the air molecules losing progressively more density. Racing fuel for street riding is one unwieldy option since it is not readily

available and is a hassle all the way round. The performance mechanic will recognize this heat expansion problem and take steps to rectify it by cooling the compressed air after it leaves the discharge side of the turbo-supercharger before it enters the intake manifold. Therefore, a denser, highly oxygenated compressed air enters the combustion chamber to do its thing. Intercooling is primarily an efficiency prerequisite. Intercooling does double-duty by actually removing heat that causes detonation. Furthermore, an intercooler will safely allow use of higher boost levels in conjunction with fully advanced ignition timing on readily available pump gas. The bottom line is that intercooling allows maximum performance with maximum engine component protection.

## Normalized Turbochargers

There are two basic types of turbocharging and each affects the rider use of the throttle. The normalized turbo regulates the amount of compressed air inducted into the

engine.

Wastegate

Excess air bleeds away automatically through a valve called a This turbo normalization places no throttle limitation on take-off since excess air dissipates.

Non-normalized Turbochargers

Non-normalized turbochargers have a fixed valve or Since it is not automatic, it is possible to feed too much air into the engine causing a condition called Over-boosting will cause a reduction in engine longevity or even cause engine failure in extreme circumstances. The throttle cannot turn fully from a standing start without possible dire consequences with a non-normalized turbo. Therefore, it is obvious to choose a normalized turbo so that excess air can bleed away via the

wastegate.

## Turbocharger Cams

The goal with turbocharging is to have as little exhaust backpressure as possible relative to the chosen boost of forced induction. Turbochargers create an exhaust restriction that raises exhaust gas pressure. The pressure of exiting exhaust gases, encouraged by an open exhaust valve and a rapidly rising piston creates exiting high exhaust gas velocities that spins the turbo turbine. It is very important to match the turbocharger and engine combination so that boost pressure is higher than the exhaust pressure particularly near the peak torque of the powerband.

The pressure differential of boost relative to exhaust pressure will approach 1:1 much higher in the powerband

near peak horsepower. Exhaust pressure will increase beyond boost pressure at or shortly after peak horsepower.

I repeat that exhaust backpressure needs to be less than boost. With these conditions met, the turbocharger cam selection will be similar to that of the supercharger.

## Consequences of Wrong Turbo Sizing

However, some manufacturers will make the turbo size a little smaller than desirable to match the engine to reduce turbo lag. This will result in too much exhaust backpressure relative to boost. Turbo tuners may make the mistake of increasing boost to compensate but this will start to affect turbo efficiency and the gains that ensue.

The correct procedure in this case is to make cam-timing changes that reduce valve overlap further. If both valves are

open at the same time, the boost that stacks the compressed incoming air will cause reversion back out the valves.

First, improper matching of turbo to engine most easily and efficiently corrects by replacing with a properly matched size turbo.

However, an alternative to altering cam timing is to increase oxygen content into the combustion chamber. Oxygen is king not necessarily the boost increase. Power has everything to do with oxygen. Porting and polishing the head ports, will be a big assist in increasing flow. The correct intercooler will also help immensely as cooler air intake is a denser oxygenated intake.

Of course, using the proper size turbo in conjunction with a turbo cam that will use little overlap with lower duration

figures and a relatively higher lift and proper flowing heads with a good intercooler is the optimal choice leading to maximum turbo power. Again, a super-turbocharger cam will approximate a stock emissions control configuration.

## Intake Valve Opening

Our cam choice will have the exhaust valve closing near or at piston top-dead-center. This creates a situation where cylinder pressure will be greater than intake pressure. The piston needs to descend in its cylinder on the following intake stroke with the intake valve closed to reduce this residual exhaust pressure past an equalization point to prevent reversion. Now we have a strange situation that would never occur with a naturally aspirated engine whereby the intake valve opens after the intake stroke begins and not before it. Boost needs to coordinate with intake valve opening.

Therefore, intake valve opening should occur anywhere from to of crankshaft rotation ATDC (after top dead center) depending on amount of boost. A street turbo setting of close to 15-psi boost will broach closer to ATDC with race applications (30 psi boost), approaching the ATDC configurations. Lower boost applications will open the intake valve closer to TDC. A better throttle response will occur if the exhaust valve stays open a little after TDC in this situation.

Increasing boost will necessitate delaying intake valve opening to prevent reversion of gases. In a perfect world, the intake valve should open when boost pressure is equal to cylinder pressure. Negative overlap occurs when the exhaust valve closes (TDC) but the intake valve delays opening into its stroke by 20 to 35-degrees thereby creating negative overlap of 20-to –35 degrees. Of course, this will cause reversion losses.

Correct Turbo Sizing

When the turbo matches closely with the engine configuration, there will still be pumping losses but the pumping gains will more than make up for this. No matter the valve timing, some losses will occur. However, it is crucial that exhaust pressure does not override intake boost pressure. If exhaust pressure overrides intake boost then intake-fuel-charge reversion will occur.

The reverse is also true with higher boost, which can easily cause exhaust pumping out the exhaust valve if open. Valve timing is also crucial here because blowing the intake charge out the exhaust will cause ignition and burning of the charge in the header pipe. This will raise temperatures substantially around the exhaust valves and in the turbo.

As with any hi-performance application, there is always a balancing act finding the best combination of factors to produce maximum safe power. The LSA or lobe separation angle should not be less than 106-degrees and may approach as high as 114-degrees. Wider LSA's decrease valve-overlap while narrower LSA's will increase valve overlap.

Valve Lift

Another situation arises with little or no valve overlap and exacerbates with negative overlap. The mechanical lag time that it takes to get valves actually opening far enough to allow increasing flow is a reason for opening valves early and closing them late.

Super-turbochargers place high flow demands on low valve lifts. They also require less duration and valve overlap to

prevent reversion back into the intake and flow-through the exhaust because of the forces of the higher boost compressed charge.

These forced induction engines also need higher lift valves. How do we get the valves opening sooner and higher in this situation?

Increase the Rocker Arm Ratios

On a turbocharger, intake-cam-opening can delay as much as 35-degrees ATDC. Peak piston velocity will occur somewhere around 70-degrees ATDC. Naturally aspirated engines, open the intake valve well before top-dead-center (BTDC). There are some serious time restrictions placed on the time the intake valve gets to fully open especially in light of when the piston vacuum capability of drawing in the intake

charge maximizes. Furthermore, the compressed intake charge boost is not able to enter the engine well into the intake cycle.

Increasing the stock Twin Cam rocker arm ratios of 1.625 will open the valves faster and further. Ultima (www.jirehcycles.com), supplies a 1.675 rocker arm ratio, while Crane Cams (www.cranecams.com), supplies rocker arms with a 1.750 ratio.

If I use a stock TC96 camshaft as an example, I can demonstrate the effect on valve lift that increasing the rocker arm ratio will have. Valve lift is the result of multiplying the actual cam lobe lift of the camshaft by the rocker arm ratio. Knuckleheads had a rocker arm ratio of 1:1 so that cam lobe lift equaled valve lift. The stock Evolution and Twin Cam engines have a rocker arm ratio of 1.625:1.

The TC96 cam-lobe-lift centers around .290” depending on

the year plus whether it is the intake or exhaust lobe.

Therefore if I multiply the cam-lobe-lift of .290” by the rocker arm ratio of 1.625:1, the resultant valve lift will be .471”.

- If multiplying the TC96 cam lobe lift of .290” by the Ultima rocker arm ratio of 1.675:1, the resultant valve lift increases to .488”.

- Finally, if multiplying the TC96 cam lobe lift of .290” by the Crane rocker arm ratio of 1.750:1, the resultant valve lift increases to .508”.

Stock 1.625

TC96 valve lift is .290” cam lobe lift x rocker arm ratio of

1.625 = .471” valve lift

Ultima 1.675

Stock TC96 valve lift is .290” cam lobe lift x rocker arm ratio of 1.675 = .488” valve lift

Crane 1.750

Stock TC96 valve lift is .290” cam lobe lift x rocker arm ratio of 1.750 = .508” valve lift

Longer ratio rocker arms typically move the power band down increasing engine torque and horsepower primarily in the midrange. Higher lift always assists in take-off torque. However, the goal in the case of forced induction is to get the valve open faster to overcome mechanical lag time and the constraining effects of reducing valve overlap. A stacked-up compressed boost intake charge must contend with low valve

lifts that result particularly with negative valve overlap. It is not that the valves cannot open fully but they cannot do it in time for maximum intake piston speed and other engine efficiency demands.

## Hydraulic Lifter Rollers

The problem with Harley-Davidson rollers on the hydraulic lifters that follow the cam lobes on their rotational rounds is that they cannot accelerate valve opening as fast as an automotive flat tappet. However, after about 25-degrees cam rotation, the hydraulic roller catches up to the speed of a flat tappet. After this, the hydraulic increases speed over the tappet.

The advantage of hydraulic rollers is that they suffer less frictional horsepower loss. As I discussed under Cams for

hydraulic rollers cope more easily with the increase necessary in valve-spring-seat-pressures to control valve actions.

Cams for turbochargers and superchargers, for the most part need low overlap, low duration, higher lift and wider lobe separation angles. Please refer to the earlier section, Supercharger Cams for more information.

## EFI Twin Cam Turbocharger

Trask Performance (www.turboyourharley.com) has a bolt on turbocharger kit for 2002 and later EFI and carbureted Twin Cam FLT series Baggers. Trask claims that with correct installation and proper tuning from an otherwise stock bike it will produce between 115 and 130 HP and 115 to 130 ft-lbs torque with 8 lbs of boost on 91 Octane pump gas.

Each kit supplies with a highly modified Garrett GT Series turbocharger, billet plenum with built-in velocity stack, billet breathers, billet cam cover, ceramic or chrome pipes, hi-flow injectors and a performance Barnett clutch spring. They also offer an optional intercooler kit. This kit requires one of the following, a Dynojet Power Commander (www.powercommander.com), a Screamin' Eagle Race Tuner (www.harley-davidson.com), or Zipper's Thundermax (www.zippersperformance.com). A CD is included in the kit that supplies all required base maps. Please see Chapter IX: Fuel Delivery, sections, Engine Management Software, and Power and Screamin' Eagle Race Tuner and Direct Link for more information on this subject.

Optional finishes are available for all kits as well as an intercooler option. I recommend to always using an intercooler.

Twin Cam Softail and Dyna Turbocharger

This system works on 2000 and later Softails and 1999 and later Dynas. Carbureted or injected, this kit is easy to install and makes great power with 115 HP and 120 ft-lbs torque on a stock engine with 8 lbs of boost on 91-octane pump gas. Trask also offer an optional intercooler kit.

EFI Sportster Turbocharger

Trask also has a bolt on turbocharger kits for 2007 to present EFI Sportster models. They claim 113.3 bhp at 6200 rpm with 110 ft-lbs of torque from an otherwise stock 2007 XL 1200R. Trask supplies the same components as with the Bagger Turbocharger.

V-Rod Turbocharger

This new Trask turbo kit is specifically for all 2002 to 2008

V-Rod models. Claimed ratings are 160 HP with 110 ft-lbs torque. Trask also includes a custom high performance clutch along with ignition and fuel maps.

The only difference between the Nightrod Special kit and the Standard V-Rod kit is the coating finishes. The Nightrod kits are in Black on Black finishes to match the black theme. Otherwise, all components are the same. 4

## Choosing a Supercharger or Turbocharger

How does the rider decide between a supercharger and a turbocharger? This is like discussing religion. In general, superchargers have one major advantage over turbochargers. Superchargers do not suffer lag, which is how much time passes between the rider twisting the throttle and the engine's response.

## Reducing Turbo Lag

As I have said, turbochargers suffer from turbo lag because it takes a few seconds before the exhaust gases reach a velocity that is sufficient to drive the impeller/turbine. One way to decrease turbo lag is to reduce the inertia of the rotating parts, mainly by reducing their weight. This allows the turbine and compressor to accelerate quickly, and start providing boost earlier.

## The Use of Wasted Energy

Theoretically, a turbocharger is more efficient because it is using the wasted energy in the exhaust stream for its power source.

## Exhaust Backpressure

On the other hand, a turbocharger causes some amount of backpressure in the exhaust system and tends to provide less-boost until the engine is running at higher rpms.

## Reliability

Will a turbo or supercharger shorten the life of a Twin Cam (or any other type) engine or drivetrain?

All other factors being equal, it is the type of bike riding that directly affects engine life whether it has forced induction or not. In general, turbocharging or supercharging a Twin Cam will not shorten its longevity. I can only point to the longstanding reliability of car and truck engines utilizing turbochargers or superchargers.

Centrifugal supercharging, which generates boost in line with engine rpm is the easiest on engine components. However, Roots and Twin-screw blowers low rpm boost can place additional strain on the engine and drivetrain.

Normal riding, which includes not giving it a handful until there has been a rolling start will not increase wear on the drivetrain. Superchargers will not increase powertrain wear with normal riding. However, when racing the additional torque provided by turbo-supercharging will place additional load on the drivetrain. It is the same deal as how the owner treats a stock Harley.

Engine and powertrain impact minimizes when the boost increases in tandem with engine rpm, as is the case with Centrifugal supercharging and turbocharging.

## Choose a Reliable Product and a Competent Performance Mechanic

As always, patronize someone with the knowledge and skill required. I already know that after this book is publishes some hapless souls will come to me to fix problems of forced induction units purchased and installed elsewhere. That is not to say that there are not fine mechanics and shops out there; it is just that there are more crummy ones than good ones.

Your wrench will also need knowledge of compression ratios, how to achieve them and know about the types of camshafts that work best with forced induction. This knowledge, in many ways, is counter to what makes a naturally aspirated engine fast.

## Turbocharger Cool-down

The one caveat for many turbochargers is that they have a cool-down period after running but this is common knowledge for competent mechanics that must explain this.

## There is One Constant: Augmenting Airflow

In naturally aspirated Twin Cams, porting and polishing heads, increasing volumetric efficiency via physics principles such as scavenging at higher rpms, enhancing swept volume via big bore, and/or stroking the engine are some traditional ways to increase performance. Forced induction, by compressing air into the cylinders above ambient atmospheric pressure via turbo-supercharging magnifies volumetric efficiencies to increase power dramatically. There is one constant. All the ways of making an engine faster revolve exclusively around getting more air into the engine.

# Twin Chargers

A Twin Charger is a relatively new term where an engine integrates both a supercharger and a turbocharger. As a logical mind might deduce, both forms of forced induction comes into their own during the varying rpm range. At lower rpms where there is not much, exhaust pressure to drive the turbocharger impellers, the supercharger stacks the air into the cylinders. At higher rpms as exhaust pressure naturally builds, the turbocharger has more influence by compressing the air into the cylinders in higher volumes, thereby increasing performance.

The Volkswagen Golf GT is such a vehicle that incorporates a supercharger and a turbocharger on its direct-injection, small-displacement 1390 cc (1.4 liters), engine that produces 170 horsepower and 177 ft-lbs of torque. The

relatively high compression ratio of 10.0:1 and the low-end boost of the supercharger effectively expunges turbo lag.

- The supercharger boost rules this engine up to about 2400 rpm.

- From 2400 rpm to about 3500 rpm, both the turbo and supercharger contribute to power.

- Above 3500 rpm, an electromagnetic clutch disengages the supercharger.

- The turbocharger works exclusively above 3500 rpm.

- The engine-management-system ECU computer seamlessly controls the transition between the supercharger and turbocharger.

The European GT that is not currently available in the USA accelerates to about 100 kph (62 mph), in just under 8-seconds. Top end is about 218 kph (135 mph), where this vehicle obtains a remarkable 16.6 kilometers per liter or 39 miles per U.S. gallon.

## Supercharging, Turbocharging, and Nitrous Oxide

I am now going to proceed into discussing nitrous oxide. There are benefits to using nitrous oxide with turbocharger or supercharger applications. In turbo applications, turbo lag eliminates with the addition of a nitrous system.

Furthermore, both turbochargers and superchargers compress the incoming air, thus heating it. The heating air expands losing some of its oxygenation potential. With nitrous

injection of nitrous, its cooling effect reduces intake charge temperatures by C F), or more. Therefore, the air contracts and becomes dense increasing its oxygenation capabilities.

With nitrous oxide in combination with supercharging, turbocharging or twin charging, it is possible to increase boost further to add even more power to the equation.

## Nitrous Oxide

The ongoing theme for creating more power is to increase air capacity in the cylinders. The oxygen in the air is the key component.

### Natural Aspiration

A Twin Cam engine will ingest only about .066 liters of gasoline and 3000 liters of air-per-minute when running at 4000 rpm. Cylinder fill does not reach 100% at all rpm ranges in the Twin Cam engine. This is normal of all H-D engines. However, it further constrains in the Twin Cam because of the bottleneck created by inadequately flowing heads. Many are satisfied with creating 100% cylinder fill. However, there are many ways to achieve much higher percentages of fill. We can do this naturally with the principles of scavenging whereby up to 130% cylinder fill can attain in the higher rpm ranges.

Another simple way is to create a ram flow affect. If the air manifold that connects the carburetor or EFI body to the intake ports of the heads sweeps outward and forward then the carburetor/EFI throat will be situated in the way of air rushing past the moving motorcycle. Air will force or ram into the cylinder heads in much greater volume and speed.

Stacking more air can also accomplish by cooling the

intake charge as the air will be more dense therefore providing more oxygen. If the air intake is hot, the air will expand becoming less dense therefore providing less oxygen. Higher cylinder fills are also possible by artificial means by stacking the air via techniques such as supercharging or turbocharging forced induction.

However, skilled mechanics will always use their knowledge to burn more of what is already there, thus increasing the efficiency of the internal combustion Twin Cam engine. Therefore, the efficiency of how well air and fuel is mixed is a subsidiary but extremely important way to create more power since more of what is already there will burn.

One might think that more gas should be the key to more power and it can be. The bottom line is that to put in more fuel, the delivery of around 12 to 15-times more air per unit

of fuel depending on the needs of the engine. Creating more power is all about utilizing more air.

## The Legendary Nitrous Oxide

I am going to discuss the release of more oxygen into the fuel via the near instant performance and legendary nitrous oxide. The introduction of nitrous oxide (N2O), more correctly known as dinitrogen monoxide, into an engine delivers more oxygen during combustion. Nitrous oxide is an oxygen-bearing compound. Its chemical formula is N2O. This means that each nitrous oxide molecule has two nitrogen atoms and one oxygen atom.

Nitrous oxide is often and incorrectly known by the acronym NOS. However this is the name of the company, Nitrous Oxide Systems (www.holley.com), which is the probably the largest marketer of nitrous oxide injection

systems for the internal combustion engine; be it a car, motorcycle or whatever.

## Laughing Gas

Nitrous oxide is a colorless, non-flammable gas with a slightly sweet taste and odor. Dinitrogen monoxide has use in dentistry and various types of surgeries because of its anesthetic and analgesic qualities. It is not toxic or irritating and when inhaled in small quantities can produce mild hysteria and good times. Thus, we have the nickname laughing gas.

When inhaled in pure form it can cause fainting or in rare cases death by asphyxiation because of a lack of oxygen in the blood. This seeming contradiction of this oxygenate explains by the fact that the oxygen in nitrous oxide is not available to

the body at atmospheric temperatures and pressure.

Atmosphere in a Bottle

The composition of the air that naturally aspirated motors breathe in is approximately 75-parts nitrogen, 4% miscellaneous gases and only 21% oxygen. Therefore, only 21% of the air ingested in an engine is of use for combustion purposes.

- Nitrous has a higher percentage of oxygen in the neighborhood of 36-percent.

- The nitrogen has no effect in combustion at all, but it does absorb heat.

- N2O is 50% denser than air at the same pressure.

- Another way to look at this is that there is about 21% oxygen in a cubic foot of air.

- With the addition of nitrous and heat, this composition broaches 50-percent more than double that of air; hence the colloquialism, atmosphere in a

Splitting N2O into Oxygen and Nitrogen

When nitrous oxide heats to about degrees F C), it splits into oxygen and nitrogen. Therefore, if there is more oxygen, then the performance mechanic can introduce more fuel, allowing the same engine to produce more power. Nitrous oxide is one of the simplest ways to provide a significant torque/horsepower boost to any internal combustion gasoline engine.

## Nitrous Cools the Combustion Chamber

Furthermore, nitrous oxide has a cooling effect that improves performance even more.

When the pressurized nitrous leaves the bottle it changes from its liquid state to a gas when it sprays into the engine. When it vaporizes, nitrous oxide provides a significant cooling effect on the intake air.

- N2O lowers the intake air temperature. Therefore, the air's density increases, allowing more oxygen to pack the combustion chamber to combine with an increase in fuel delivery.

- Nitrous oxide boils at Celsius; it will begin to boil as soon as it injects, because the temperature in this area is

greater than this.

- During this process, it absorbs heat and this process can cause a 40-degrees or so drop in inlet manifold air temperature

- A 10-degree Celsius drop in temperature can add 2% power to an engine.

It is important to understand that the expanding heat-pressure that drives the piston down on the power stroke is a result of the gasoline and not the nitrous. If adding nitrous without extra gas, there will not be extra power. The only result will be an increase in the rate the gas will burn. This will lead to a lean AFR and the very real danger of detonation.

Use Nitrous Sparingly

The amount to run an engine continuously cannot store on a motorcycle.

Rule of Thumb: Nitrous Oxide Volume Usage Formula

0.8 lbs N2O x 10 seconds = 100 horsepower

For example if a Twin Cam engine can produce 100 horsepower then it will require 0.8 lbs of nitrous for every 10 seconds of usage

The amount of nitrous use, of course, depends on the level of power generation. It does not take long to go through a great deal of nitrous that will empty the pocketbook.

Nitrous History

Interestingly, in the discussion, earlier in this chapter in section Roots Supercharger the discovery of superchargers was by a fellow whose name was Roots about 35 years before the first cars. Their initial purpose was to ventilate mineshafts.

Since, I am on the subject of mineshafts, the guy that invented a measurement for energy including light bulb wattage also defined horsepower. His name was James Watt (1736-1819) who delineated how much work a horse could do hauling coal out of mineshafts came up with the formula for horsepower, which is the amount of work a horse can perform in one minute. After very arbitrary calculations, James Watt decided that one horsepower equals 33,000 ft-lbs of work per minute.

Then in 1900 in Germany, a fellow named Daimler

(Daimler-Chrysler) installed the first Roots Supercharger on a car. Germans also began the use of nitrous oxide in internal combustion engines during the onset of World War II. Their nitrous system dubbed GM-1 had extensive use in Luftwaffe reconnaissance and fighter planes.

The Royal Air Force in Great Britain soon followed suit.

As the lore goes, America shied from nitrous usage in their warplanes because they felt it too dangerous. However, some of these non-nitrous equipped planes had their own boost through supercharging.

For the real speed freaks out there, the obvious question arises of whether supercharging or turbocharging can combine with nitrous oxide? Yes, it can with a resultant stacked and oxygenated air charge to mix a correspondingly greater amount of gas.

## Is Nitrous Safe?

Nitrous has always had the mythology of danger around it. Like a drug, some tend to abuse it. The abusers created the bad rap. With inappropriate use, engines and powertrains will suffer. Used sparingly and knowledgeably for fun or otherwise, nitrous is very safe giving definite advantage to the user.

In airplanes, the ongoing theme of more air for efficiency and performance illustrates once again that by adding nitrous oxide to the intake charge will wonderfully compensate for air density reduction and thus less oxygen at high altitudes.

## NASCAR

America rediscovered nitrous in the 1950's when a famous stock car racer named Smokey Yunick won races while using Nitrous. However, NASCAR put short shrift to Smokey when they outlawed nitrous use. This did not prevent nitrous usage totally, as there have been many accusations and actual scandals involving cheating with nitrous.

## Drag Racing

In the 1970's, nitrous became popular in drag racing. The street scene soon followed with hot-rodders. Motorcyclists followed a short while later.

Nitrous oxide in a Twin Cam is the equivalent of a jet's afterburner; it has use for short periods-of-time for an extra burst of power.

## How Nitrous Works

When nitrous introduces into the intake tract of the Twin Cam engine, it enters into the combustion chamber. On the compression stroke, as the rising piston compacts the air, gas and nitrous, a very oxygen rich mixture results as the temperature reaches approximately Fahrenheit C), thereby breaking down the nitrous oxide into nitrogen and oxygen atoms.

This has a considerable supercharging effect, if I may use that term.

Another way to look at it is as a compression increase. The dynamic cylinder pressure in the engine dramatically increases during combustion with nitrous. Therefore, a much higher amount of burnable fuel can add to the nitrous mixture with the resultant power gain. The gain is very substantial with upwards of a minimum 30% increase.

## Detonation

Earlier, I state that cylinder pressure increases. This can have the negative effect of encouraging the dreaded engine destroyer detonation. This is where nitrous gets its bad rap; not from the detonation but from the mechanic that does not control it.

Excess heat raises concerns with detonation, which causes the fuel to ignite spontaneously without an assist from the spark plugs sparking. Detonation and the modern Harley Twin Cam, and Sportsters for that matter, seem to be partners. This is largely due to emissions controls. The heat generation required to limit pollutants on stock machines is due to overly lean air-to-fuel ratios. However, other factors can also cause detonation including low octane gas or too much compression.

Detonation is the nexus of nitrous harming engines. Again as usual, there is truth to this but it comes from insufficient knowledge. Retarding ignition timing controls detonation on nitrous engines during its use.

Another cause of damage from nitrous is using too much. The ensuing cylinder pressure increase may damage pistons and or rings or even head gasket failure.

No two cylinders ever run exactly alike, which is why it is always necessary to check both spark plugs. Nitrous is different from natural aspiration in that its use cleans the spark plugs to new condition.

## Dry, Wet, and Direct Port Nitrous Delivery

The specific nitrous kits for an air-cooled engine like the

Twin Cam must address the challenge of detonation control with a nitrous oxide system.

Liquid nitrous storage in the bottle enters the supply lines as a flowing liquid. Electrical solenoids control flow out the purge valve if there is one, for efficiency in clearing lines of gaseous mixtures and air. A solenoid, which is an electrical coil that acts as a valve also, allows feed of the nitrous as an atomizing liquid spray into the intake manifold via a nozzle, nozzles, or a plate. The solenoids require wiring and a switch(s) to activate them.

Proportional Delivery is Crucial

Excess nitrous produces too much oxygen relative to the gas. Too much air creates a lean condition, creating heat, which of course leads to detonation.

Conversely, if there is too much gas relative to the nitrous intake then the resultant fat spots will rob power and create pollutants. The chemically correct nitrous to gasoline ratio is 9.65:1.

Nitrous must install correctly and set up proportionately both to be effective as a power producing oxygenater and to protect the longevity of the Twin Cam engine.

- Small doses of nitrous will equal a very safe 30% power gain.

- The rule of thumb with engines utilizing nitrous is that if desiring gains of over 40% then it will require upgrading engine parts.

This is very true with a Shovelhead engine. It is safer with

an Evo engine since its construction is better than a Shovel.

- However, the Twin Cam, in my opinion, can safely have gains of up to 50%.

The Twin Cam engine components excepting those in the timing compartment, which I replace with gear drives, are extremely strong. All other H-D connecting rod and crank pin components pale when comparing to the very beefy Twin Cam ones. The pinion bearings in the Twin Cam are humongous. The hypereutectic piston construction is superb in the Twin Cam. The metallurgy and casting of Twin Cam engine casings is the best. Therefore, we have a very superior engine here in which, to use nitrous for performance gains.

Dry Nitrous Delivery

A dry nitrous oxide delivery system does not mix the fuel

and nitrous automatically. Detonation is an ongoing concern with a dry nitrous system because EFI delivers a specific amount of fuel to combine with varying amounts of nitrous. Therefore, the danger is always an inappropriate amount of nitrous relative to fuel delivery, which will result in a lean condition that is the breeding ground for detonation.

It is common to run dry nitrous delivery into EFI manifolds. Nitrous injects as a spray at the throttle body while the gas injects at the intake ports. Additional gas injection through the injectors is necessary to richen the lean nitrous oxygenation through bigger injectors, a fuel modifier, and/or increasing injector on-time or pulse width.

The advantage of a dry system is that gas cannot puddle in the manifold where it must negotiate a turn into each cylinder. This is detrimental first to fuel flow that should be in

the cylinder not liquefying in the manifold. More importantly, puddling is a breeding ground for an explosion that creates much inefficiency by blowing the induction system off the bike. It may not be this serious but it is still not good.

- The nitrous will not split into nitrogen and oxygen until cylinder temperatures reach Fahrenheit C).

- However, an additional benefit to nitrous is that it will cool the intake charge by up to Fahrenheit C), which means the air will be dense and able to carry that much more oxygen capability.

Nitrogen acts to buffer, or dampen the increased cylinder pressures helping to control the combustion process.

- Dry nitrous kits are excellent for first-time nitrous users.

- A dry delivery kit is excellent for normally aspirated combinations that have a return-style-fuel-system like EFI.

- I do not recommend using a dry delivery system on forced induction turbocharged or supercharged engines.

- Use a wet nitrous delivery system for forced induction applications.

Wet Nitrous System

A wet nitrous oxide delivery structure distributes nitrous oxide and additional fuel in the correct proportion into the combustion chamber. More correctly, wet nitrous systematically introduces correct proportions of nitrous and atomizing gas into the incoming air stream. More fuel adds to the air-fuel-ratio because of the smaller amounts of oxygen

present in the air stream but also the larger oxygenated content of the nitrous.

- Wet systems are very popular because they are simple and usually easy to install.

- Nitrous oxide and fuel, each with its own inlet, inject through the same wet system nozzle that has one outlet.

- There are two solenoids, with two supply lines to the nozzle inlet, one for nitrous and one for fuel.

The dry system nozzle is a singular affair for nitrous only.

- Carbureted wet systems will make the upper intake wet with fuel via a plate that injects nitrous and fuel.

- EFI wet systems places the nozzle a few inches away from the throttle body.

Wet nitrous oxide works well as a standalone oxygenater or in conjunction with turbocharging or supercharging.

Direct Port Nitrous Delivery

The advantage of direct port nitrous delivery, which is a different type of wet system, is the ability to tune each Twin Cam cylinder separately. This can be very important as dry or wet nitrous delivery may result in a rich condition in the front Twin Cam cylinder and a lean one in the rear cylinder or vice versa since this is a form of forced induction. Direct port nitrous and gasoline inject via a common nozzle mounting directly into the individual ports in the intake manifold, usually at a close point to the cylinder head.

Direct port delivery allows metering of the nitrous and gas into each intake port separately. This system will usually result in the most torque/horsepower delivery but it is more complicated than dry or wet nitrous.

Racing will usually dictate the use of direct port nitrous injection.

Plate Nitrous Delivery

A plate device installs between the carburetor and the intake manifold that is similar to a carburetor spacer except that it injects nitrous and gas into the engine. A nitrous plate has separate spray one for fuel and one for nitrous that span across the intake manifold opening.

They spray a fine mist of fuel and liquid nitrous directly

into the intake manifold.

Edelbrock Progressive Nitrous Controller

Nitrous injector pulse width can increase or decrease nitrous delivery by its The same volume of nitrous exits the jet for any given time-period. Increasing or decreasing pulse width on-time will affect the volume of nitrous.

The Edelbrock progressive controller (part # 71900), allows the rider to control the amount of nitrous over a period-of-time by providing a pulse width modulation to the solenoid.

Riding with Nitrous

Riding a Nitrous equipped Twin Cam requires some common sense such as no hole shots. Those who do hole shots for effect will suffer the same fate as those that smoke their tires; they may need a truck to get home. In order to save the powertrain from undue wear or breakage caused by sudden power it is prudent at least to have a rolling start before injecting nitrous oxide.

Edelbrock supplies an rpm adjustable nitrous control module that allows the mechanic and/or rider to preset at what rpm the nitrous will activate. It cannot set lower than 2000 rpm for good reason.

- The rpm at which the system activates can select in 200-rpm increments from a range of 2000 rpm to 5000 rpm.

- The second adjustment switch controls the rpm limit at which the system shuts down. This adjustment is in 125-rpm

increments starting at 5250 rpm and ending at 7125 rpm.

It is even safer to save nitrous until in higher gears.

Using a larger bottle rather than a small streamlined one is beneficial to giving more stable nitrous pressure. It is a falsehood that a bigger bottle gives more torque/horsepower. However, the more stable pressure provided by a bigger bottle will still probably win the race because a smaller bottle may not be able to always provide this stability for as long. It also follows that it is smart to race with a full bottle for the same reason.

An upgraded ignition is not a necessity the same as with a stock Twin Cam.

## The Nitrous Purge Valve

As with any riding, there are tricks to learn that may give an edge over others with less knowledge and experience.

The use of a purge valve will allow the serious rider to purge all of the gaseous nitrous oxide from the connecting supply line between the bottle and injector before usage. This will give a harder hit. If the concentrate liquid nitrous oxide dilutes with weaker gas in the unpurged connecting supply line, the nitrous oxide hit to the combustion chamber will be softer. A purged supply line will result in a more efficient running engine that all other factors being equal will also win the race.

When opening the nitrous bottle valve, nitrous oxide flows up the line to a solenoid. It is important to purge this line beforehand. A nitrous purge valve allows clearing the delivery

lines as a precursor to immediate power delivery from the nitrous bottle. The actual purging requires another solenoid that allows the line to clear. Allow purging until a consistent fog of nitrous exits the solenoid line.

Purging will also allow an over-pressurized bottle to vent until to the recommended pressure. Follow the nitrous bottle manufacturer's instructions; otherwise keep the bottle between 850-950 psi. It is beneficial to develop a nitrous habit of purging the supply line after a nitrous run and then turning the bottle off.

In this way nitrous is always immediately available

## The Forgotten Tuning Aid: Spark Plugs

Who would ever think that spark plug heat ranges could assist in heat reduction inside the combustion chamber?

Spark Plugs and Gasoline for Nitrous

Colder spark plugs are a good idea for use with nitrous oxide.

There is a simple and common methodology to determine whether a spark plug is colder or hotter than another is. The numbering system in the spark plugs part number that appears on the side of the plug will indicate this; the lower the number, the hotter the spark plug; the higher the number, the colder it is.

After reading this complete section on spark plugs, the reader will be able to tell if a plug is hotter or colder than another one, by the length of the tapered insulator nose. A general rule of thumb is to use spark plugs that are cooler by about one point cooler.

Always do plug-readings to ensure the correct plug is in use. It is advisable to experiment with two points cooler to see if the engine can tolerate this change because of better heat removal. Use these with the highest-octane gasoline available from the pump.

- Do a short-nitrous run for the initial plug check.

- Then confirm with another short burst.

- A Twin Cam nitrous plug will be white to off-white without blistering.

- Blistering on a white background is too lean and dangerous.

The condition of concern, as always is detonation, which is very bad for a naturally aspirated engine, is very much worse on a forced induction nitrous one.

- A tan color is okay but broaching a rich condition.

- Brown coloring on the electrode is rich.

Detonation evidences by one or more of the following, black dots, and/or a greenish tinge and of course, a burned electrode. Check both plugs, all the time. The front cylinder air-shrouds the rear from the wind stream on the air-cooled Twin Cam and therefore, the rear cylinder runs hotter.

Furthermore, forward momentum of the bike in acceleration and a reverse momentum as the bike decelerate or brakes will affect the flow of gas, nitrous and air differently

to each cylinder. This may well cause temporary leanness in one cylinder relative to the other.

Spark Plug 101

Twin Cams and Evolution Sportster use 12mm thread spark plugs while earlier. Shovelhead and Evolutions use a 14mm thread.

All 1975 to present Big Twin, which includes the Twin Cams plus all Evolutions including Sportsters from 1986 to present use long reach plugs. Ironhead Sportsters and 1974 and earlier Shovelheads, Panheads, and Knuckleheads use short reach spark plugs. The difference is the number of threads on the spark plug that screws into the same number of threads in their mating cylinder heads. Short reach spark plugs have maybe one-half the threads of a long reach plug.

They are prone to cross-threading and possible thread stripping the cylinder head inserts. Long reach plugs were a great idea and resolved many problems.

The outside spark plug white insulator is usually ribbed but some are smooth. The ribbing's purpose is to create a longer path between the top terminal, where the spark plug wire cap fits over, down to the metal base where the plug screws into the head. The design purpose is to reduce chances of high voltage arcing down the outside of the spark plug. In actual practice, I am not aware of smooth insulators allowing any outside arcing.

The spark plugs heat range lists, usually numerically as part of the part number on the side of the insulator. However, there is no uniformity in part numbers between manufacturers. Furthermore, some manufacturers like Harley-

Davidson have one part number on the spark plug and a second different one in the parts manual although they do relate to each other.

Heat range has nothing to do with how hot or cold the spark is but with heat removal, which I will discuss in the next section.

There are two basic types of plug, resistor, and non-resistor. There will be an R somewhere in the spark plug part number to indicate a resistor plug. It is okay to run a resistor plug in any Harley-Davidson. However, a resistor plug is very necessary in any Harley that uses an electronic ignition, which, of course is every Twin Cam, Evolution and Shovelhead going back to 1978. Resistor plugs prevent radio frequency interference (RFI), in the computerized ignitions. A non-resistor plug will allow interference that will disrupt the engine management system ECU. The very first electronic ignitions shut the bikes off if near an airport whenever the air

traffic controllers were using certain radio frequencies to communicate with the planes. The use of resistor spark plug wires is also necessary with resistor spark plugs. Mechanical points-condenser ignitions use non-resistor spark plugs and spark plug wires.

Please see Volume VI: How it all for much more information on spark plugs

Spark Plugs Remove Heat

A little known facet of the lowly spark plug is to remove heat from the combustion chamber. The heat range of the spark plug defines its ability to dissipate heat. Part of the spark plugs responsibilities, in addition to firing a spark, is to remove heat from the combustion chamber. This accomplishes by channeling the heat through the insulator material and

metal housing to both the cylinder head and surrounding air for heat dissipation.

One full point in heat range reduction will remove approximately F to F C to C), from the combustion chamber. A cold plug acts as a conduit for heat to guide out of the combustion chamber. However, it cannot be too cold to allow fouling of its electrode.

The heat range must be hot enough to burn off gas and carbon deposits, which will prevent the spark from arcing across the electrodes. Gas fouling will leave a black fluffy residue covering the electrodes while oil fouling leaves black shiny and scaly deposits. Fouling prevents or semi-prevents ignition of the fuel charge causing stalling, lousy acceleration and poor overall performance and efficiency. Therefore, nitrous applications should use the coldest plug possible without risking fouling because of higher than normal temperatures developing because of higher compression,

forced induction and other factors.

However, if too cold, idle quality and low-speed motorcycle operation quality is at risk because of electrode fouling. A spark plug heat range that is too hot for a particular application may assist in preignition and detonation that may cause power loss and engine damage.

The Heat Range of a Spark Plug

The heat-dissipation capability of a spark plug determines by the length of the insulator nose, which is the tapered portion of the spark plug that carries the center electrode into the combustion chamber.

The hooked outer electrode strap, appropriately known as the ground strap extends over the center electrode. The space

between the electrode and ground strap is the spark plug gap, which is adjustable by bending the ground strap away from or closer to the fixed center electrode.

Combustion chamber heat transfers through the center electrode enclosed by the insulator nose up through the body of the plug to the cylinder head and surrounding air. Therefore, hot plugs have a longer insulator nose. Thus, hotter plugs expose more insulator-nose-surface-area to the combustion gases.

Colder heat range plugs have a shorter insulator nose, minimizing the surface area exposing to the hot combustion gases. Cold plugs transfer heat faster through the electrode to the head and outside cooling air.

Forced induction will often create higher heat in the

combustion chamber so it is an important detail to run a colder spark plug for the purpose of detonation reduction. Sometimes, the length of the insulator nose protrudes into the combustion chamber too far allowing the ground strap of the electrode to heat up excessively with the faster and more intense nitrous burn. In effect, the ground strap cannot dissipate the heat and will become a glow-plug causing detonation.

The detonation in this case is not a result of spark plug of a too-hot heat range although many mechanics will mistakenly assume this. It is necessary to shorten the path from the spark plug ground strap to the plug base. This accomplishes by replacing the plug with one with a shorter ground strap. A colder plug with a shorter insulator nose may still have too long a ground strap since the two may or may not coordinate. The length of the ground strap is the problem not the length of the insulator nose.

Pointing the arcing spark between the center electrode and the grounding strap towards the center of the combustion chamber is one of many incidental sneaky speed secrets. The thorough performance mechanic will take the time to carry out all the incremental little things that may add up to winning or losing a race.

It only makes sense that the ground strap will shroud the spark from the combustion chamber if it positions wrongly after threading into the spark plug hole. Positioning is the luck of the draw. Who is to know where the thread will begin on either the spark plug male insert or the head's female threaded insert? Thus, the spark plug compression washer thickness is a key to positioning the electrodes inside the combustion chamber towards the action. This metal 2-layer

single washer is more correctly a gasket or a compression gasket.

Another more laborious and expensive method is to buy a quantity of spark plugs and keep trying them until finding one where the electrodes position correctly towards the fuel charge. Mark the spark plug exterior with a magic marker or similar non-graphite marker to indicate where the open electrode gap is pointing. Using a lead marker like a pencil can cause problems since it can conduct electricity; although unlikely, this may short or arc the incoming voltage.

1. Install the plug to torque specification using Loctite Medium Strength Thread locker on the threads.

2. Torque for 2007 TC96 touring models is 12 to 18 ft-lbs (16-24 Nm). Always consult the manual for correct specifications. I torque to the middle of the acceptable range.

3. Spark plug gap, in this case, .038” to .043” depends more on fuel mixture. A wider gap will ignite a leaner fuel mixture whereas a slightly richer mixture does not need as much gap. See below for optimal gapping for nitrous oxide.

4. Observe where the marked position on the plug’s exterior points.

5. If the marking indicates the electrodes are pointing to the center or intake side of the center of the combustion chamber, all is well.

6. If not, which is more likely, there are spark plug indexing washers of varying thickness that substitute for the compression washer in order to juxtapose the electrodes towards the center-intake side of the combustion chamber.

7. Do not combine washers to get the desired thickness; only use one per spark plug.

Indexing is a small thing that probably won't affect a street bike much but it is a culmination of many small procedures that will make the difference.

## Power Tuning Nitrous

If not tuning each cylinder separately, always tune for the leanest cylinder, which is usually the rear one. Actually I can say that almost always this is the case on a Twin Cam particularly closed loop EFI ones. Shovelheads, on the other hand sometimes ran richer on the rear.

### Spark Plug Gap

The Twin Cam spark gap for natural aspiration is .038” to .043”. For nitrous, it is a tuning secret to close the plug gap approximately .010-inch tighter than stock. The only concern with this is possibly shielding the spark. The exception to this tuning ploy is when using a performance ignition modifier is to follow that manufacturer’s tuning instructions including spark plug gap if any. Otherwise, with a performance ignition return to the Factory spark gap settings.

Do not increase spark gap setting beyond Harley specifications unless instructed by the performance ignition specifications. The concern here is that if the gap opens too far and the spark fails to make the jump, the cylinder will misfire.

Maintaining the Correct Air-to-Fuel Ratio

It is obviously necessary to tune nitrous oxide and gasoline for delivery into the combustion chamber and maintain the correct air-to-fuel ratio (AFR). As I discuss earlier under Detonation in this chapter, nitrous causes cylinder pressure to rise. Increases in cylinder pressure also increase its corollary, heat. Excess heat may lead to detonation. Follow religiously, the manufacturer's instructions, as they will invariable deal with heat issues.

- Utilize spark plugs that are at least one range colder than recommended for the naturally aspirated Twin Cam engine.

- In review a single one-range heat range reduction will remove approximately F to F C to C), from the combustion chamber.

- Always do plug readings on both cylinders after a short

nitrous run when tuning or making other changes.

- Always maintain the engine in top condition.

- Do not use the nitrous if there are engine problematic symptoms.

Gasoline Octane

Always use the highest-octane pump gasoline possible. Lower octane ratings will mean that the tuner should retard ignition timing. Be aware that the longer the nitrous squirt the higher the load on the engine's internal components.

Upgrade Engine Components

If using nitrous frequently for longer bursts, the engine components such as pistons, rings, connecting rods and bottom end bearings probably need upgrading.

- Twin Cam hypereutectic pistons, massive connecting rods, wrist pins, and 3-hole crank pin along with more surface area on bottom end bearings meet the bill.

- Evolution eutectic pistons, with smaller con rods, wrist pins and single-hole crank pin combining with less bottom end bearing area does not bode well for sustained high nitrous usage without upgrades.

Hypereutectic or forged piston upgrades with beefier aftermarket connecting rod packages and a 3-hole crank pin may be necessary.

- Shovelheads will require at least the above for sure.

Carburetor Nitrous Tuning

Smaller carburetion is better in a nitrous application. There will be plenty of oxygenation, so air will not be as much of a concern.

Ignition

I have already talked extensively about nitrous and spark plugs.

- Smaller nitrous kits will work well with the stock ignition system found on Twin Cams and Evolutions.

- Shovelheads will need an upgrade to an aftermarket

electronic performance ignition.

- Big-inch engines will require a hotter ignition.

- Nitrous requires less advance timing because the oxygenated fuel charge will burn very quickly.

Advancing ignition means the spark plug will ignite the fuel charge with the piston further from TDC when traveling upwards on the compression stroke allowing the fuel charge more time to burn. Retarding ignition timing means the spark plug will ignite the fuel charge with the piston closer to TDC when traveling upwards on the compression stroke allowing the fuel charge less time to burn. Moving or changing advance-retard ignition timing will mean changing when the spark plug fires relative to piston location in the cylinder in degrees of crankshaft rotation on the compression stroke preceding the power stroke.

- Ignition advance will increase engine power output but invites the increasing possibility of detonation.

- Retarding ignition timing will decrease engine power output but is very effective at controlling detonation.

As I said, the nitrous oxygenation allows the fuel to burn very fast.

- Therefore, ignition retard is the timing adjustment for nitrous applications.

Generally, it is acceptable to tune ignition timing about 2.5-degrees crankshaft rotation for about every 50-horsepower of nitrous oxide up to 5-degrees closer to TDC. Do not retard

timing past TDC into the power stroke. Too much retard ignition timing will cause the exhaust headers to glow red. This results from the creation of excessive heat traveling into the exhaust headers with a consequential power loss.

Twin Cam EFI engines may not need an ignition adjustment to the timing as the electronic ignition module (ECM) may automatically compensate for any minor adjustment requirements.

1. After adjusting the ignition timing, monitor to see if detonation occurs.

2. If so, retard one degree at a time until detonation disappears.

3. If running fine, increase (advance), ignition-timing one-degree at a time until detonation occurs, then retard one-

degree.

## Nitrous and High Compression

Nitrous is suitable for both low and high compression engines. However, as always it is crucial to maintain the correct balance of nitrous and fuel enrichment.

With higher compression ratios, it is also important to retard ignition timing sufficiently to control any possibility of detonation.

As compression increases, so does the need for higher-octane gasoline. It will reach a point that race fuel may be necessary. Some gasoline nitrous race engines approach unbelievable 15:1 compression ratios.

# Nitrous Jetting

Nitrous jets fit into the end of the nitrous nozzle for multiple power settings with an accompanying fuel jet in wet systems. These metering jets similar to carburetor jets manage the power level by controlling nitrous spray. Jet sizes measure in thousandths-of-an-inch. Therefore, a number 40-jet has a passage diameter of .040”.

When encountering problems with engine misfire or detonation, the procedure is always to reduce the size of the nitrous jet first.

Power results from burning fuel pressure. Nitrous only allows the fuel to burn.

Fuel addition will cool a naturally aspirated engine.

Adding more fuel in the case of nitrous is like adding gas to a fire. Therefore, adding fuel will not cool down nitrous as more of it will now burn.

- If the mechanic increases nitrous oxide jet sizing, the engine will lean out.

- A decrease in jet sizing will enrich the air-fuel-ratio.

This is opposite to natural aspiration physics-principles with which, the reader will be more familiar.

Necessary Reduction of Nitrous Jet Sizing

As I have said, detonation symptoms include tiny silver or black speck deposits on the porcelain and/or a greenish tint on the plugs.

1. A burned electrode on a sparkplug is a clear sign of detonation

If present, in addition to retarding the ignition also reduce the nitrous jet size.

2. Furthermore, if the spark plug ground strap that folds over the center electrode to form the spark plug gap exhibits a bluish-rainbow coloring also reduce the nitrous jet size.

3. If the ground strap indicates signs of melting, once again reduce the nitrous jet size but also change to a spark plug to one with a shorter and thicker ground strap.

The rider may not encounter much of this stuff when using

kits supplied by NOS or Edelbrock that tailor to a particular engine like the Twin Cam.

These kits have preset controls and safeguards to prevent the malady of detonation caused by excessive nitrous delivery.

Increase Nitrous Jet Sizing

If plug readings are good and there is no audible detonation,

- Increase nitrous jet size, one-step at a time.
- Keep doing this patiently until there is evidence of detonation.
- Once detonation occurs, decrease the jet one size down.

- Always take the time and make the effort to do plug readings on both cylinders.

- If not tuning both cylinders separately as is often the case with Harley-Davidson, always tune the leanest cylinder!

# Nitrous Bottle Heating for Performance

Nitrous oxide (nitrous) does not store in its gaseous state.

## Nitrous Oxide Storage

N2O stores in an external container after compressing until it liquefies. Liquid nitrous oxide is very compact and dense. However, the storage container or bottle is not aesthetically clean to many car buffs. Nevertheless, in the wacky world of Harley, bulky and even ugly can transform into cool and beautiful. The storage of nitrous oxide under necessary high pressure uses an aluminum or carbon fiber cylinder.

The cylinders or bottles range in weight from about 2.5 lbs to 20 lbs. I can give no meaningful typical usage statistics to

the reader. There are too many variables including the length of nitrous run. However, a 10-lb bottle typically might give five runs with a 125-horsepower kit. The same 10 lb bottle may last up to ten-quarter mile runs with a 75-horsepower kit.

I do not recommend spraying nitrous more than 15-seconds maximum per run. As bottle, pressure drops so does the delivering torque/horsepower. Nitrous bottles need to maintain between 900 psi and 1000-psi pressure to achieve optimal torque/horsepower. This figure varies with bottle-to-bottle and manufacturer-to-manufacturer.

Of course, the first prerequisite is to keep the bottle full to maintain full pressure. If bottle pressure is too low, a rich fuel charge may result. While this may not be harmful to the engine as if a lean condition would, it defeats the purpose of having nitrous oxide forced induction in the first place.

Conversely, it is necessary to cool the nitrous bottle in very hot temperatures because bottle pressure will increase steadily to create lean conditions in the combustion chamber posing a danger of detonation and possible engine damage. Nitrous gauges will indicate too cold or too hot conditions resulting in less than desirable bottle pressure conditions. Gauges are a tracking indicator of bottle psi; the only accurate way to measure the fullness of a nitrous bottle is to weigh it.

Nitrous achieves maximum power when the nitrous bottle is at least 20% full or more. Surging and power-drop will indicate a below 20% capacity.

Example; if the bottle temperature is C, bottle pressure is only 760 psi; if the bottle temperature is C F), bottle pressure

will only be 590 psi. A thermostatically controlled bottle heater will raise bottle temperature and thus bottle pressure to between 900 psi and 1000 psi for optimal performance nitrous spray.

Ambient outside temperatures of to C to F) will allow for best power potential of nitrous kits

My next temperature statement has nothing to do with bottle temperature. However, nitrous also acts much like an intercooler on supercharged or turbocharged engines in that it reduces intake air/gas temperatures by up to Celsius F).

## Nitrous Oxide Cams

It is fine to use the normal stock emissions cams with nitrous oxide applications for two reasons.

1. The first is that most of the time nitrous is not in use with wide-open throttle (WOT).

2. The second is emissions cams have very low exhaust-intake overlap, which coincides with nitrous delivery needs.

Therefore, a normal performance cam may also be okay as long as it is a low overlap one with more exhaust lobe duration. There are special nitrous cams available for competitive racing but they are not appropriate for the street. It is best to follow the nitrous manufacturer's recommendations for your model bike as weight and gear will have an influence.

## Edelbrock Nitrous Systems for Harley-Davidson

The venerable Edelbrock has a longstanding excellent reputation for performance products and nitrous oxide

systems are no exception.

I quote, “All Edelbrock Nitrous Systems are dyno-tested and calibrated to deliver accurate and safe horsepower with even distribution to each cylinder. Our nitrous calibrations are determined through dyno-testing, not simple mathematical formulas, which do not take into account real-world variables. Dyno proven to produce 35+ HP increase at the push of a button, Edelbrock Nitrous Systems and components are the quickest and easiest way to get large horsepower increases with a minimum of engine modifications and expense. Edelbrock Nitrous Systems have proven to be an effective power adder for carbureted Harley’s and are NOW AVAILABLE for EFI Applications as well. We have designed an industry first Radial Discharge plate for EFI applications. This high-tech radial discharge design offers unobstructed airflow

within the intake runner minimizing the risk of air/fuel ratio changes by the bikes ECU…Edelbrock's new radial discharge N20 plate for EFI Harley's bolts between the manifold and throttle body for easy installation, just like our current line of carbureted H-D plates. This is the first for H-D's that does not utilize a spray bar but rather sprays nitrous and fuel from the walls of the plate. It was designed this way due to EFI systems being extremely sensitive to changes within the runners. If a spray bar were present within the runner, the EFI system would need to be re-calibrated to accommodate for this runner change. This new plate design leaves the runner passage open and unobstructed to eliminate, or at least minimize, changes to the EFI systems map…Edelbrock RPM Nitrous plates are available in many configurations to exactly match your induction system...The ALL-NEW EFI plate kit is designed with a radial discharge for unobstructed airflow in EFI applications. 2004 and later models also require inductive RPM sensor #71937." 5

Edelbrock Performer Nitrous kits include a choice of a 12-ounce bottle or a 2-lb bottle, nitrous and fuel solenoids; aircraft braided steel lines, nitrous and fuel fitting filters, stainless steel jets and the all-important instructions.

Edelbrock Nitrous Kits

◊ 2004 to present Twin Cams require an Edelbrock Inductive RPM Sensor (part number 71937), for tachometer signal pick-up. All 2004 to present Harley-Davidson's get their rpm signal from the data bus, which makes it more complicated to pick-up a tachometer signal. The Edelbrock inductive rpm sensor allows an accurate reference of the engine rpm for components such as the Edelbrock Nitrous System or aftermarket tachometers.

1 Accessible Technologies, Inc. Website, (www.procharger.com), Motorcycle; Overview; Intercooled Supercharging ProCharger .

2 Ibid.

3 Ibid.

4 Trask Performance. Website, (www.turboyourharley.com). Page, Turbo Systems. Modified by email.

5 Edelbrock. Website, (www.edelbrock.com). Page, Motorcycles. Section, Nitrous Systems.

6 Ibid.

# Chapter XIV

## The Future

Design basing on past traditional nostalgia is on the endangered species list.

The year 2010 is a watershed year for many vehicle manufacturers with Harley-Davidson's nostalgic design being particularly vulnerable.

### Say Goodbye to our Beloved V-Twin

Please bear with me as I go through some necessary

terminology and statistics.

Year 2006 to 2009 motorcycles above 280cc referred to as Class III in EPA literature have difficult but not impossible to meet exhaust emissions standards of no more than 12.0 grams per vehicle kilometer (g/km) of carbon monoxide (CO).

One kilometer is a metric measurement that equals .621 miles or about ⅝-of-a-mile.

Year 2010 emission standards, thankfully for the survival of Harley-Davidson as we know it, do not reduce this amount.

Year 2006-2009 Class III motorcycles, which again include all Harley-Davidson's, have a limit of 1.4 grams per vehicle kilometer of hydrocarbons (HC), plus oxides of nitrogen (NOX).

Oxides of Nitrogen define as the sum of the nitric oxide and nitrogen dioxide contained in a gas sample as if the nitric oxide were in the form of nitrogen dioxide.

Gooblydegook for sure but the bottom line is that this is bad stuff and is polluting our environment.

However, the kicker is that the 2006 to 2009 limit of 1.4 g/km of hydrocarbons plus oxides of nitrogen will reduce to 0.8 g/km for 2010 and later.

How Can Present Twin Cam Design Survive?

The term later really means until the emissions dictates become even more stringent. Once technologically meeting one goal for mass production then the next ever more restrictive goal is set.

Therefore, I can now ask the rhetorical question. How can present design no matter how progressive survive future Environmental Protection Agency constraints?

I will not broach the subject of alternative power replacing oil leading to a very new type engine or a hybrid one because the world is not economically able to extract itself from this interwoven product any time soon.

However, I suppose a hydrogen fuel cell's flat cellular structure could design into a V-Twin shaped shell to satisfy our needs of the past. Surely, the fuel cells one emission, drinkable water, is an attractive alternative.

Perhaps a battery-operated Harley-Davidson will be the future. A V-Twin shape cover can sit over and around the battery. Please do not think these examples are absurd.

A few more technological breakthroughs and this will become reality.

Oh, did I leave out solar energy, which is currently growing at the rate of 50% per year?

The immediate reality for Harley is to play catch-up with existing technology that has been around and used for decades by other manufacturers like the Japanese. Of course, Harley has tried new technologies on the V-Rod and most recently the more radical departure from tradition, the 2008 1125R Buell.

However, those governments pushing for a global treaty on greenhouse gas emissions in the world 2-week conference in Bali in Dec. 2007 are progressively putting more pressure on

the world to adopt ever-increasing stricter controls beyond than are dictated by the EPA for 2010. The Bali conference set an aggressive timetable to negotiate a global agreement on post-2012 climate change targets and endorsed the International Panel on Climate Change view that dramatic action is necessary to avert an environmental crisis.

Businesses that include the automotive industry and thus Harley-Davidson can expect new demands to reduce greenhouse emissions.

Canada, for example, grudgingly accepted broad targets to reduce emissions by 2020 by between 25% and 40% from 1990 levels. In addition, Canada is in a bit of a pickle since it is a major supplier of oil to the USA with the Canadian Alberta tar sands. Extracting oil from the tar sands takes an awful lot of polluting energy. Only building a nuclear reactor with its cleaner energy can defray the green house emissions in the Canadian far north.

The pressure to cut deeper will only grow particularly if the United States embraces climate change action under successive administrations. However, even if the USA goes its own way, American manufacturers like H-D that wish to export to Canada and Europe will have to comply if they wish to keep their global market since the Bali agreement covers 38 countries.

In a surprise announcement, George Bush announces in July 2008 that the USA will reduce greenhouse gas emissions in half by 2050. My opinion is that this is a feel-good announcement. However, what do I know since power politics is well beyond my scope?

Some think the Bali Accord was not about environmental emission reduction at all but a discussion about the world

economy. This is what makes pollution control so difficult. Do we save the environment or do we save jobs and a current way of life. Do we think about us or do we thing about our grandchildren. These are the real questions. They are difficult.

How does a 2-cylinder air-cooled engine survive this? The short answer is that it does not. Although, a 2-cylinder, awkward firing, pushrod operated engine can survive a little longer with a conciliatory water jacket, overhead cams eliminating the valve train and a catalytic converter but it will still eventually be taken out by environmental concerns.

Okay, so now if the cylinder configuration changes to 56 or 60-degrees or whatever…yes, the engine's survival will prolong it a little.

The addition of multi-cylinders will prolong this

eventuality again.

Future Engines will not Power by Oil

Therefore, the stopgap-engine-design measures will prolong the life of the V-Twin, albeit with drastic changes.

The long answer is that future new engines will not power by Big Oil.

Moreover, here is the problem. In my opinion, Big Oil lobbies relentlessly with mega-dollars. Furthermore, some of the most powerful politicians in the world make their money off oil. These are the very people that must make decisions to replace oil as an energy source.

What else can I say?

# Willie G. Solution for the Radiator

Willie G. has had much to do with radiator design over the years at Harley-Davidson. We can thank him for the inoffensive look of the V-Rod radiator. Nevertheless, the V-Rod is a radical departure from traditionalist H-D design. It gains slow but steady acceptance from the riding public because of its speed.

However, this was not Willie G.'s first foray into radiator design. His original design on the little known Harley-Davidson Nova is radical, functional and out-of-sight. The Factory may make a very smooth move indeed by incorporating its patented Nova radiator designs on traditional looking water-cooled Harley-Davidson's that are in development as I write.

# The Little Known AMF Story: Saving the Hogs Bacon

A little known American Machine and Foundry story is that of the very secretive Nova Project. For H-D riders of the day, AMF was derisively famous for manufacturing bowling balls amongst other recreational products. However, AMF also experimented extensively with a new ultra-progressive design motorcycle.

Over three decades ago in 1976, AMF/Harley-Davidson began development of new two, four, and 6-cylinder water-cooled motorcycles whose technology was certainly equal to if not potentially ahead of the Japanese at that time. The projected launch date for the Nova was during 1981.

AMF also began development of the Evolution circa 1977.

The maligned AMF certainly deserves praise and respect for its efforts to save the Hog's bacon.

Probably both projects would have gone forward in tandem if AMF were not bought out. The buyout of AMF/ Harley-Davidson left the new owners low on financing. They made a choice between the producing the Nova and the Evolution. The rest is history.

## The Nova Radiator

Reportedly, Willie G. is repulsed by the thought of a radiator out front on the Nova. Even the water-cooled Nova cylinders had finning to approximate the air-cooled look of traditional Harley's just as future water-cooled Twin Cams probably will. Lore has it that Willie G. took control of radiator design and relegated its ugliness to the rear of the motorcycle placing it horizontally under the seat.

How could this design possibly work? Well it does and there is still a patent on it and the air channeling techniques that the AMF employees like Willie G. developed.

A fan underneath the radiator creates a pulling vacuum to draw air through the radiator and back out to the rear of the Nova. Two big stylistic air scoops facing forward into the wind replace the traditional gas tank. They channeled air back into a plenum chamber above the radiator to draw through its coils into the fan-created vacuum.

To be sure, a smooth move on Willie G's part as well as being very functional.

Willie G.'s work many decades ago may reach fruition in the near future.

Generate more Power while cutting Emissions

In fairness, Harley-Davidson must adhere to government regulations as difficult as they may be for a two cylinder, air-cooled pushrod operated engine.

Some cheat the system to gain much desired performance gains by removing or modifying emissions controls. Authorities have overlooked and in some cases tolerated these indiscretions for decades now. The California Air Resources Board (CARB), has been actively prosecutiing many bike shops, both aftermarket and dealerships, for removing emissions components like exhaust systems, air breathers, fuel induction, camshafts and ignition modules for years now. Some convicted shops are very prominent ones, instantly recognizeable by any Harley afficionado.

# Loud Pipes Hasten the Process

The window is closing rapidly. Abuses from the loud-pipe crowd are accelerating the process. Shaking the windows of hospitals and schools with noise is not conducive to ongoing tolerance.

I love the sound of straight pipes but I like many thinking riders learn to feather the throttle in situations that will offend others. It saves on costly ticket infractions also. It is ironic that in most cases, straight pipes cause a loss of power and especially low-end torque.

The future is developing performance options that also meet government mandates. Some think this not possible. I am not one of them.

Performance pipes have always made less noise than straight pipes. These exhausts are also quietening further with progressive companies applying technology to the noise issue. EFI allows for potentially huge power gains.

Precise fuel delivery on demand both helps reduce emissions and increase power. The challenge is to meet noise and exhaust emissions at the same time as increasing horsepower and torque.

Bob Wood's innovative belt-helical cam gear drive does exactly this. This is all possible. Emissions controls do not have to equate with power decreases.

New ideas, adapting proven automotive technology and developing new technology are the solutions.

Harley is making small changes continuously that will extend its air-cooled engines lifespans. While the jury is still out on the parent-material-bearings, the timing chain hydraulic adjusters will enforce stricter cam timing. A simple Factory change from spring-loaded timing chain adjusters to more consistent hydraulic ones to resolve adjuster pad breakdown have at the same given a performance improvement of improved cam timing accuracy albeit not as good as a gear drive but a step in the right direction.

## Lighter Harley's

The impact of reducing vehicle weight will affect cars more than Harley-Davidson's but will be a driving force nonetheless. U.S. fuel economy rules require automakers to reach an average of 35-miles per gallon by the year 2020.

## Emissions, Once Again, Drive New Design

For Harley, this all started in the seventies with venting modifications and the ongoing attempts to control errant harmful emissions from the tailpipe.

Very dramatic changes began around 1980, with the introduction of the rear belt drive to replace the too noisy chains and sprockets to comply with lower decibel noise emissions. I mention this for two reasons; the first is that this stuff has been going on for decades now. Secondly, many of the government imposed, always-increasing lower-emissions dictates, whether they be noxious exhaust fumes, gasket leaks, oil carryover, or offensive noise from, exhaust, spur gears, and even manifold air-movement sounds have led to very positive changes.

Our bikes are faster because of more efficient cylinder

head design that progressed from hemi-head Shovels to D-head Evos and bathtub head Twin Cams that also decrease emissions. More of the fuel burns creating more power and burning up more emissions at the same time.

In 2009, this future guideline is already making waves. Reportedly, manufacturers have begun slating some heavier vehicles for removal from future product programs like a new-generation rear wheel Impala. Well, if that is the case, then I had better enjoy my Avalanche truck now.

This simple rule will turn vehicle production upside down. Look to Europe today to see the future on U.S. highways. Cars and trucks are smaller and lighter than to what we are accustomed. The trend will continue.

The reasons are simple; new vehicles that are heavier than predecessor models will not be able to reach these 2020 miles-

per-gallon dictates. In addition, the drive will be on to make existing vehicles lighter.

## The Bright Side

So, how does this affect the onslaught of future change for today's Harley-Davidson we know and love? Well, from a weight perspective, the effects will be noticeable from a variety of viewpoints. I guess the moral of the story for those where the glass is always half full as opposed to half-empty is that there is a silver lining in every cloud. There are some very real advantages to lessening Harley-Davidson weight.

## Speed

The bikes will be faster, all other factors being equal. The weight-to-power ratio will change. The same amount of power

will deliver more speed because there is less weight to muscle forward.

## Handling

Handling has the potential to improve. A lighter bike is much more nimble and responsive to rider demands. The big and bulky Ultraglide will be easier for diminutive riders to control.

The Softails, if they survive, whose frames are very heavy, will have a lot more zip around those S-bends that set wave theory into real life action…perhaps the floating sensation will even disappear.

## Braking

Increased braking control will become a boon for normal

and aggressive riders alike. I ride too aggressively as I am Type A motivated to complete tasks. Once I begin thinking and daydreaming whilst riding in the hot summer sun, the speedometer needle begins its inexorable climb.

I have become an expert at controlled, straight-on slides. I constantly find myself in situations of higher speeds surrounded by slower moving vehicles. I straighten the bike, always aware of the road surface and hammer the front brakes. The front end dives as I semi-control this action by jamming on the rear brake.

Time now measures in part-seconds. I was going to use the term mille-seconds but that is beyond the human domain and enters that of ABS braking systems.

The tires inevitably, after a mere second or two, lose their

grip under this duress and begin their slide. I ride the slide straight without wavering sideways where much danger lurks. I alternately feather front and rear brakes to approximate a crude ABS system to allow the sliding tires traction only to go into a slide again.

None of this works well on a heavy Harley ridden like a sport bike. Usually the end-result is to let off the brakes altogether and power the beast around the vehicle in front, which can be an adventure in itself as I trade problems.

A lighter Harley will increase braking prowess immeasurably from the existing situation.

## Softail on the Hit List

The Softail series are the most inefficient Harley to transfer power back through the powertrain. They are also the worst

handling Harley-Davidson. The beautiful but inept rear fork arrangement loves to wiggle sideways especially whilst navigating corners. The Softails only exist because they mimic the nostalgic, clean, and very good-looking rigid frame of yesteryear. In the world of Harley, sometimes-cool good looks are the only reasons to exist.

The reader only needs to attempt lifting a bare Softail frame to realize the concentration of weight. They are thick walled round tubing with many solid cast iron pieces. Only the strong will succeed in the lifting.

Wow, just try to lift a new style Softail Springer front-end! They are very deceptive looking. The modern Springer's first use in 1989 that continues to present day is intense in its weight.

Both the Softail frame and the Springer design is a classic Harley case of function following form. Function must adapt as best it can to the demands of style.

This is old style H-D heavy frame construction. In addition, the new style Springer weighs much more than its lighter predecessor discontinued over a half-century ago

New style is the welded together thinner sheet metal square tubular construction, which is lighter weight than the Softail methodology.

Solid cast pieces are not necessary for the most part or kept to a minimum on these frames typified by the FLT, Dyna and to a lesser extent, Sportster frames. Lowering weight would be easy by trading of the Softail frame for an approximate fifty-percent lighter rigid one.

Of course, this change is unacceptable to the modern rider that expects comfort.

Why are the frame and Springer forks so heavy? Rigidity and safe construction are the short answers. In a safe and litigious world, the Springer cannot flex like those of old. It must also cope with the sudden stopping power of a disc brake caliper and rotor not like the old style sometimes-working drum brakes with spring and cable actuated brake shoes.

Heavy steel thick walled tubing construction with chunky solid cast crowns affect a desired look at serious cost in terms of money and sacrifice to the power to weight ratio, ditto for the frames.

Does one ever wonder why the Softails are so much slower

than a Dyna when they employ the same engine, transmission, and gear ratios?

I do not think we need wonder why a Softail Springer is slower than a Softail Standard is.

## The Sweetheart of American Capitalism

Harley-Davidson is the darling of American capitalism.

Its success in the last few decades embodies the American Dream.

It is no less than pride of country and its raison d'être.

In my opinion, Harley is not accountable to the same standards that others might be and certainly not those of the

auto industry.

The practical reason is that Harley production is miniscule to that of cars and therefore not as great a threat to the environment. A motorcycle power to weight ratio is also greater than that of a car even with over-weight frames and forks. Therefore, government regulations on mandated gas mileage would not be as severe a problem for Harley-Davidson.

However, I believe the impetus will come from the consumer who will demand, needed transportation cost reduction.

Surely, increasing costs will benefit motorcycle manufacturers including Harley-Davidson as consumers partially abandon cars for inexpensive transportation. This

may be Harley's future salvation, as motivation for ownership will transfer from an aging population's diminishing nostalgic needs to one of inexpensive transportation.

I get performance calls all the time from people who want to hop-up their Harley's.

Increasingly the customer wants to know how the changes will affect gas mileage.

Oil is a decreasing resource. It costs more to get it out of the ground. Finally, there is not enough to go around, especially with China and many other Eastern countries where billions of people are going through their industrial revolutions as we did over a century ago.

As gasoline cost increases exponentially, major pressures will face Harley-Davidson. The first, of course, is to find an

alternative energy source. Say goodbye to your beloved V-Twin when this pressure becomes insurmountable.

Harley-Davidson and the Harley rider will attempt to avoid this alternative to the bitter end.

## Increasing Gas Costs to Pressure Harley-Davidson

The Motor Company at the consumer's behest will find every economy available to forestall major change. This will include new technologies. The Factory will seek minor changes in every aspect of production and efficiency to combat dramatic change.

One of these minor but very effective changes will be to increase the power-to-weight ratio. Accomplishing the raising of power relative to weight will involve a second prong.

Increase engine efficiency to gain more power per BTU of gasoline. BTU is the acronym for British Thermal Units. BTU's define the amount of heat generated per unit gasoline.

BTU Efficiency

In 1973, The United States economy requires 17,440 BTUs of energy to create one dollar of GDP (gross domestic product). In 2006 the U.S. only required half (8835 BTUs) that to create the same GDP.

This energy-conservation and efficiency movement continues unabated.

In 2006, the consumer received the same value for one tank of gas that required two tanks back in 1973. I realize no one feels this, but for value received, gasoline really costs the same. However, perception is reality and I like you do not

perceive this.

Harley is already increasing efficiency on many fronts such as cylinder head design and increasingly sophisticated engine management systems and will continue to do so with vigor.

The second prong is relatively new territory for the Motor Company, which is weight reduction to increase power-to-weight ratios. The short form to this discussion is to bid goodbye and fond farewell to the onerously heavy components like the Springer front forks and the Softail frame. The extremely heavy Softail oil tank will evolve into pressed sheet metal like those used in other models. Fatter wheels will give way to lighter narrower wheels and on it will go.

I guess aerospace technology will lead the way for Harley-

Davidson as it already is for automotive manufacturing after the obvious components, some of which like carbon fiber, I discussed above eliminate from and increase the power to weight equation.

Stronger and very much lighter materials will dominate motorcycle construction.

However, in the end, the handwriting is on the wall.

## The Throwaway Harley

A future problem will evolve that of the Cruise Drive 6-speed mainshaft or one of its four integral gears wears, it will necessitate replacement of them all since they are one piece. The same is true with the countershaft and its two integral gears.

My comment is “Here we go again.” It will become prohibitively expensive to repair the new transmissions. In my opinion, it is just one more step in assuring that Harley’s will eventually become throwaways.

Since 1903, Harley-Davidson design encouraged rebuilding almost forever on a cost-effective basis. Flathead 45 cubic-inch models served well in World War II after shipping overseas where in some cases their exposure to corrosive salt air and water and their use in desert warfare where many burned, and damaged in explosions. Yet, they still survive today. At Heavy Duty Cycles, (www.heavyduty cycles.com), we still rebuild four or five a year for their avid owners, as well as a couple of old Flathead 80 cubic-inchers from the thirties. Just try to ride a modern Harley around the Sahara for a year or two and see what the sand does to them.

Planned obsolescence defined as a calculated period of time and/or usage that a component will wear started, in some observers' opinion when H-D began experimenting in different alloys and manufacturing methods with the Evolution motor cases in 1987. The Factory replaced many thousands under warranty.

There is no disputing that the elimination of the Sportster engine guide blocks in 1991 that housed the tappets, with the introduction of the 5-speed rear belt Sportsters is an obsolescence factor. This combines with a motivation for less expensive manufacturing and mitigates with better materials.

The engine case now houses the tappets. Once worn beyond factory specifications, the engine casings will need replacement instead of the predecessor inexpensive tappet guide blocks. This will make rebuild costs prohibitive relative to the bike's resale value, especially when compared to the cost of a brand new Sportster that is rubbermounted, has EFI,

a fat rear tire, and fatter forks. It continues with the elimination of guide blocks in the 1999 Twin Cam, which will become a similar scenario over time.

Later Evolution engine casings eliminate the steel insert that houses the pressed-in left-side power-shaft end bearings. Press-fitting bearing races into relatively softer aluminum will not occur many times before wearing the casing out. In late 2002-early 2003, the Twin Cam also saw elimination of the Timken races and tapered Timken sprocket shaft bearings in favor of the same roller bearing used in the pinion side to support the flywheels.

Now, we have the new six-speed whose mainshaft is integral (one-piece), with first, second, third and fourth gears. The countershaft is integral with fifth and sixth. This leaves six of the twelve gears in the six-speed as individual gears and

serviceable in H-D vernacular. I can guarantee the reader that the rebuilding costs are going to be horrendous when compared with the twelve separate and serviceable gears with two detached stand-alone shafts, the countershaft, and mainshaft.

Planned obsolescence is a modern manufacturing principle utilized in order to ensure a manufacturers' survival. Parts including the whole vehicle have designed-in usage limits before needing replacement. A factory would not survive today building old warhorses that never became obsolete. After the marketplace saturates with everyone who wanted the product already owning one, then who remains to purchase new products to keep the manufacturer growing?

Harley's have to wear out or become obsolete after a period to ensure a rollover of buyers. Many times new

changes to a vehicle are an improvement. However, after some years, OEM (original manufacturer equipment) parts are no longer available to service out-of-date motorcycles. Without service parts, what will happen if not for a healthy aftermarket? If changes happen too quickly, there are not enough bikes of that genre for the aftermarket to profitably serve and protect.

All of this is inevitable for survival-of-the-fittest manufacturers.

## Catalytic Converters

Global Emissions Systems (www.gesi.us, near Toronto, Canada has reinvented the catalytic converter with their new GESi Technology that reduces up to 100% of Hydrocarbons (HC), up to 99%Carbon Monoxide (CO), and reducing Nitrogen Oxides (NOx) and carbon dioxide in gasoline

combustion. This new technology has use with gasoline, diesel, propane and other fuels.

Catalytic converters use metals like platinum, palladium and sometimes nickel to react with toxic exhaust gases and convert them to if not harmless, at least less dangerous substances. GESi uses slightly different metallurgy in different quantities to achieve their emissions goals.

The cost is around $1000.00 to $2000.00 U.S. per vehicle installed. The GESI converters measure about six-inches by eight-inches for a car so are very adaptable.

## The Uncrashable Motorcycle

I know it is not a Harley but the uncrashable motorcycle is almost here complements of BMW. Onboard sensors feeding

computers will detect if the bike is beyond parameters that will precipitate a crash. Computers will cause a series of chassis hydraulic arms and other pivoting components to override rider decisions to keep the bike upright preventing crashes from instability by instituting corrective balancing.

Normally, the rider will shift weight, feather brakes and attempt to steer out of a crash situation. It now gets very freaky for a veteran motorcycle rider. In order to accomplish crash prevention, handlebars may move independently of the front wheel, or wheels may turn independently via computer instruction to rectify imminent crash situations.

I wonder if this technology will lower insurance rates.

## The Aftermarket Response

Many aftermarket companies are already manufacturing

2010 compliant engines with carburetors!

These include all the majors with which the reader will be familiar.

I am italascing some telling phrases or sentences for the reader's condisderation.

I am going to quote an industry leader, S&S Cycles, Inc. (www.sscycle.com), "the availability in April 2008 of V-Series engine families certified using the California Air Resource Board (CARB) engine certification option. These engines will be certified in compliance with the pending rules and regulations of the California Certification Procedures for Highway Motorcycle Engines…To be considered for approval, the customer must follow the requirements provided with the engine pertaining to installation, maintenance and reporting

requirements…S&S customers will have the option of choosing from 50-State certified V-Series engines in 4-inch bore 113ci (V113C) and the 4 ⅛-inch bore 124ci (V113C), both will be equipped with carburetors and a carbon canister for evaporative will also be a notice explaining the MSO for the engine will not be provided until engine manufacturer receives a statement from the owner that they have read and understood the requirements for proper installation of the engine into the vehicle. Some of the information listed in the Installation Manual will include the maximum gear ratio (n/v), maximum vehicle weight, exhaust system back pressure range, maximum fuel tank capacity, evaporative system connections, and label placement. The Owner's Manual will include the emission component warranty, engine maintenance requirements, and anti-tampering 1

As great as this is, I feel it only a stopgap measure. The handwriting is on the wall; custom choopers and/or big cubic inch engines must have a necessary carbon canister for

evaporative emissions. Well, that's kind of unchopperish, now isn't it?

Furthermore, the restrictions for certification also include the maximum gear ratio, maximum vehicle weight, exhaust system backpressure range, maximum fuel tank capacity, evaporative system connections, and label placement. The Owner's Manual will include the emission component warranty, engine maintenance requirements, and anti-tampering requirements.

Well now, there goes a performance exhaust system, non-emissions cams, and a performance ignition plus perhaps some performance carburetor jetting. Moreover, there are restrictions such as gear ratio choices as well as the weight of the bike.

Then, there are the anti-tampering requirements. The only good requirement from a performance perspective is the weight limitation clause, which will increase the weight-to-power ratio in a favorable manner. Perhaps the EPA and CARB are attempting to educate all the Softail riders that performance their motorcycles with their unfavorable weight-to-power ratios.

S&S Cycle, Inc. further states in another press release about the stringent TÜV certification, which is similar to but not the same as the EPA and CARB in the European Community that covers the majority of Europe, “the successful approval of a TÜV closed loop fuel-injected T-Series EURO III engine program. The S&S T124 engine meets the current EURO III noise and exhaust emission standards and is certified for use in 2002-2006 Harley-Davidson touring motorcycles. Each engine will include an S&S VFI system, closed loop air/fuel ratio control; single bore throttle body…and the

necessary TÜV documentation for the engine. Installation of the engine will require adding oxygen sensors to the stock header pipes and the use of stock 2007 mufflers to meet EURO III emissions. The engines will be available in March 2008. S&S has also just received TÜV approval on Super E carburetor kits as replacement components for 1984-2006 Harley-Davidson big twins and 1986-2003 Sportster models. The carburetor kits include…TÜV documentation." 2

For the TÜV engines, please notice that stock header pipes, 02 sensors and stock 2007 mufflers must accompany the use of big inch stroker engines. What can I say? Big inch engines do not run very well with stock mufflers or a stock emissions engine management system utilizing oxygen sensors.

I think this is the beginning of the end. We need a new engine employing different technologies. The aftermarket will step upto the plate, I am sure as S&S has with their new X-Wedge engine.

At some point Harley-Davidson will necessarily have to develop a new technology engine, which may very well base on the V-Rod with a water jacket, different cylinder angularity and overhead cams. The Sportster will probably adapt to the new Buell 1125R engine that also water-cools with overhead cams and and a different cylinder configuration.

## The S&S X-Wedge

The X-Wedge is a completely new design V-Twin engine devised to use conservative strokes of 4.000” to 4 ⅜” combining with very safe massive big-bores of 4 ⅛” to 4 ½”. These stroke and bore combinations will produce engines from a respectable 110 cubic inches up to a massive 139 cubes. Big-bore is the way to go and very safe when designing and manufacturing engine cases for their use.

The engine development combines with a mind to complying with both air and sound emissions regulations while still producing much power.

The X-Wedge is an air-cooled V-Twin with pushrod operated overhead valves that meet the 2008 CARB and 2010 EPA regulations.

The 3-Cam

One major advantage of the H-D Twin Cam is that the dual cams lessen but not eliminate the severe pushrod-lifter angles we see in previous H-D engines. The X-Wedge carries this theme a step further. S&S almost eliminates any pushrod-lifter angles in favor of the much more efficient straight up and down configuration.

This new engine uses three camshafts. The center camshaft services the two intake valves, which have always been the straightest up and down pushrods on any of the engines discussed here. There is a separate exhaust cam on either side of the dual intake cam.

In this way, S&S straightens pushrod angularity.

Serpentine Belt Drive

The advent of the Twin Cam engine eliminates the use of Evolution timing case gear drives with supposedly quieter chain drives. In the X-Wedge, three serpentine belts drive the large base circle-cams thus eliminating gears, chains and sprockets. A flat rubber drive belt turns multiple pulleys in the timing chest. The serpentine belt gains its descriptive name by snaking around the various pulleys. A spring-loaded pulley

maintains tension on the serpentine belt doing away with the need to re-tension replacement belts.

Serpentine belts generally last 25% to 50% longer than conventional V-belts.

The 560 Cylinder X-Wedge

The inherent firing problems and associated vibration that is endemic to the H-D style V-Twin engines is done away with the X-Wedge, which goes to a 56-degree configuration.

Automotive-style rocker assemblies and roller tappet-lifters combine with the above for a very quiet emissions friendly engine.

Will the X-Wedge beat the pollution cops? Yes, for the present, since the engine already compies with 2010

regulations.

However, one must ask, will the X-Wedge meet subsequent CARB, EPA, and TÜV regulations as they inexorably tighten further.

1 S&S Cycle, Inc. Website, (www.sscycle.com). Press Release, La Crosse, Wisconsin. February 7, 2008.

2 S&S Cycle, Inc. Website, (www.sscycle.com). Press Release, La Crosse, Wisconsin. February 11, 2008.

# Chapter XV

## Twin Cam Performance Summary

With this genre engine, much of the Twin Cam motor is a bulletproof platform for the near future of Harley-Davidson. Designed with durability and reliability in mind, the engineers were expecting a minimum 100,000 miles before a rebuild.

Excepting the 1999-2006 problematic timing chest components, which the performance mechanic easily fixes, the longevity expectations are likely understated.

Many times, my writing necessarily focuses on the problems and fixes associated with Harley-Davidson

motorcycles. This may create a climate of negativity after a period, simply because the reader may lose sight of the wonderful aspects of the Twin Cam engine relative to what has gone before.

## The Twin Cam Platform for Performance

The Twin Cam bore is 3 ¾" for both the TC88 and the TC96, which is a big increase over the Evo's 3 ½" while the stroke decreases to 4.000" from the Evo 4 ¼" on the TC88 and to 4 ⅜" on the TC96. The Twin Cam stock bore increase of .250" (¼"), over the Evolution does not add any stress to engine operation. The reason is heat producing piston speed remains the same and does not increase. Bigger bore is the safer way to more cubic inches. However, the stroke in the Twin Cam decreases a full .250" from the Evolution in the TC88 and increases .125" over the Evo in the TC96. Piston

speed slows in the TC88 relative to the Evo and marginally increases with the TC96.

As I have discussed, the improvements in casting qualities, parts design, parts excellence, engineering superiority, metallurgy, improved and new materials, and quality control all join to allow the safe use of mega-bores and longer strokes that were not possible in the past.

This is big news for the performance set. The S&S 124 cubic inch behemoth with a secure stroke of 4 ⅝" (4.625"), and a mega bore of 4 ⅛" (4.125"), builds easily and safely using the superb stock Twin Cam engine casings!

They are fast and they hold up because of two overriding factors, the quality of the S&S parts and the quality, strength and durability of the stock Twin Cam engine casings.

Now, the industry is building bigger engines using aftermarket engine casings with wider cylinder spigot-hole walls allowing the safe use of even bigger-bore cylinders such as the R&R Cycles Inc. with a 4 ⅝” stroke and a 4.600” bore.

JIMS has a 131” Twin Cam using an even more reliable 4.500” (4 ½”), stroke and 4.3125” (4

I believe will see much bigger inch Twin Cams in the near future. The Twin Cam engine development, designs to run stock for much, much longer than any Big Twin of the past does. A few serious glitches like the timing chest components require salvation from the aftermarket. A performance obstruction with low head airflow also easily manages with porting and polishing techniques or with head replacement for bigger engines.

This is why H-D engineers call this engine a platform. The serious performance mechanic can use this structurally solid base for wild torque and horsepower increases in a variety of safe ways. Therefore, if the rider's heart desires, the Twin Cam staging area for more power is available and it is fast, resilient, and hardwearing.

## The Twin Cam Weakness

Noise pollution and exhaust emission restrictions drove Twin Cam design.

Controlling exhaust emissions creates heat and therefore a cylinder detonation concern. This ever-increasingly serious issue will exacerbate with the new 2010 EPA guidelines, surely following with dictates that are more restrictive as time goes on.

However, it was lessening noise that primarily drove Twin Cam design. The spur straight-cut timing gears used in previous models like the predecessor Evolutions and all that went before them were far too noisy to pass auditory emissions testing. Noise generates from three main sources; the exhaust, gears, and unbelievably air rushing through the intake manifold. The Motor Company attempts to muffle air intake noise and lessen gear noise to preserve the potatoe-potatoe exhaust noise that H-D aficionados love.

The replacement timing chain apparatus in the Twin Cam is problematic in the early Twin Cams and may develop in new problems with late Twin Cams. In my opinion, this preservation attempt of one of the most marketable facets has been unsuccessful within the restrictions the EPA places on Harley-Davidson.

Stoichiometric Air Fuel Ratio

A new strain is increasingly beginning to aggravate the smooth operation of the newer Twin Cams concerning burning higher percentages of pollutants in the combustion chamber. This aggravating and painful tendonitis has as its cause of excess heat and the symptom of potentially harmful detonation. This will result in actual harm in the new CVO 110 is because the excess heat derives from ratios exceeding the Stoichiometric air-fuel-ratio for gasoline of 14.7:1.

This is a very lean burning and therefore, heat generating closed loop air-fuel-mixture. Closed loop EFI, which we have on the newer Twin Cams, Sportsters, and V-Rod's means that while cruising the air fuel ratio is Stoichiometric 14.7:1. The Stoichiometric Ratio defines as the optimum air-fuel-ratio where all the available oxygen in the air combines with all the available fuel for burning in the combustion chamber.

We will get maximum HP and torque with an AFR less than 14.7:1.

Ovaling of Bearing Holes

A law of physics, Moment Arm Force explains the potential 2006 Dyna and all 2007 to present timing chest problem in the making. The parent material bearings in the cam-support-plate may oval over time and usage because the cam drive sprockets are closer to the support plate than the other camshaft end.

If the sprocket of the camshaft were in the middle, Moment Arm Force would distribute evenly and ovaling would not be an issue.

Many automobiles use parent material bearings very successfully but the camshaft drive sprocket is in the middle negating the ovaling effects of Moment Arm Force.

CARB, EPA, TÜV Cometh

The ever-restrictive emissions controls now result in 2010 compliant aftermarket engines utilizing a carbon canister for evaporative emissions. They must also include a maximum gear ratio, maximum vehicle weight, exhaust system backpressure range, maximum fuel tank capacity, evaporative system connections, and label placement. The Owner's Manual includes the anti-tampering requirements.

Well now, there goes a performance exhaust system, non-emissions cams, and a performance ignition plus perhaps some performance carburetor jetting. Moreover, there are

restrictions such as gear ratio choices as well as the weight of the bike.

Then, there are the anti-tampering requirements. The stringent TÜV certification, the EPA and CARB equivalent in Europe, aftermarket performance engines must use stock header pipes, 02 sensors and stock 2007 mufflers must accompany the use of big inch stroker engines.

Big inch engines do not run very well with stock mufflers or a stock emissions engine management system utilizing oxygen sensors.

Conclusion

The Twin Cam engine cannot survive for many more years in its present arrangement and configuration.

Liquid cooling will surely be a near-future change, as will a move away from the design to or perhaps and the elimination of the pushrod valve train and moving the cams into the cylinder heads from the lower timing compartment.

Engine Management will become ever more sophisticated as will combustion chamber design.

However, no matter how much the Motor Company attempts to utilize new technology with an antiquated base design; the present engine configuration cannot survive.

# Chapter XVI

## 2009 Model Changes

Superficially, there does not appear to be any major changes with the 2009 Twin Cam models.

Nothing could be further from the truth.

There are many new and sometimes-subtle changes that the rider and/or mechanic should not take for granted.

If working on any of the new Harley-Davidson's for the past few years do not assume that specifications are not

constantly changing.

Always consult the Harley-Davidson manual.

I constantly find innocuous changes.

One reason is that H-D is wisely keeping traditional appearance a priority.

## The Twin Cam Touring Frame

The FLT touring frame has not appreciably changed in 29-years since its introduction in 1980. It has now supported three different engines, each faster and more powerful than the one before.

A CVO 110 has almost double the horsepower and torque

of the original Shovelhead engines that had basically the same FLT frame fitment.

The new frame is stronger and increases the GVWR (gross vehicle weight rating) by 100 lbs (45.5 kg), to 1360 lbs (617 kg). Furthermore, this assists in increasing cargo carrying capacity by 15 lbs, 5 lbs for the tour box and 5 lbs for each saddlebag.

The previous FLT frame consists of 90 different castings, stampings, and bent pieces of tubing.

There are no bent pieces of tubing on the 2009 FLT-series frames. There is only straight tubing that is larger in diameter and thus stronger. Cast and forged modular node-tube joints hold the new frame together. There is about half the parts in the new frame and only half the welds. The new frame

backbone skeleton builds around a single spar that begins as 2-stamped pieces, welded together.

I hate to say this but eliminating the human element with coordinated motion robotic MIG welding ensures accuracy, welding quality and chassis alignment.

Two-Piece Frame

The frame is now two pieces. There is a separate tail section. This is primarily for possible future design flexibility on the part of the Motor Company. Changing rear tire size or rear mounting of component parts like the battery box, fender, saddlebags, and tourpak is now a simple procedure whereby only a small separate part of the frame needs resign instead of the whole enchilada.

Customizers will have fun with this also, as the

possibilities expand exponentially. The rear end of the new touring bikes will become a customizing hotbed of new ideas and looks.

The frame is reportedly 20% stiffer for better handling at speed and in the corners. I can report that the new frame handles a great deal better.

Geometry stays much the same with a steering rake of However, trail increases .7” to 6.9”, again for more comfortable handling.

The rear fork is much beefier and two-piece construction. The manufacturing lingo is the fork is from a deep-drawn shell with a forged pivot section.

Engine Isolation-mounting System

Handling and vibration control improves once again by expanding the previous three-mount-system to four. The single center upfront isolation mount replaces with two side-mounted vertical isolation supports. One mounts at the base of each frame downtube and tie together with a single link.

Now each of the four corners the engine have isolation-supports instead of a shaky triangle. The new supports are supposed to last the life of the motorcycle. This is great but I must say that I have heard this type of claim before notably for the early disastrous 1999 to 2006 Twin Cam spring-loaded timing chain tensioners. Logically, the new isolation mounting system will last much longer since there is extra support that does not allow wide swings in engine vibration especially at idle.

The improvement effect is most noticeable at idle. The

engine does not shake as it has from 1980 right up until 2008.

## Shocks

There are now dual adjustable air shocks replacing dual non-adjustable air shocks.

## Internal Engine and External Rider Cooling

Harley-Davidson did not make any one comprehensive fix for internal heat problems that are injurious to rear cylinders and heads, particularly on the CVO 110 or external ones that protect the rider from engine and exhaust heat-dissipation. However, there are a series of adjustments, re-designs and partial fixes that combine to provide relief internally and externally. In fairness, the one overall fix that would eliminate heat-related problems is not available to the Factory as it

violates EPA and CARB regulations. This, of course, is lowering the Stoichiometric 14,7:1 air-fuel-ratio to increase the cooling gas relative to air mixture.

Therefore, the CVO 110 continues to receive patchwork Band-Aid measures that dance around the heat issue.

EITMS (Engine Idle Temperature Management System)

The Engine Idle Temperature Management System is now rider controlled. The EITMS feature located in the stock ECM or electronic control module can turn on-or-off via the throttle grip. Once enabled, the rider can turn off fuel delivery to the rear cylinder when the bike is The rear cylinder then becomes an air pump as cooling air introduces and expels with the four engine strokes. This, of course cools off the rear cylinder that has much propensity for over-heating. Once the throttle turns for acceleration away from idle, the EITMS turns off

automatically.

The 2008½ and 2009 EITMS calibration cannot be used on early-2008 and previous years touring models.

There are new ECM fuel and ignition calibrations for 2009.

## The Oil Reservoir

Oil re-circulates more efficiently because of a baffling redesign inside the oil reservoir. Oil enters the oil pan and flows circuitously over the top of a baffle plate to the rear where oil flows down into the lower baffling to make up to nine tortured flow-turns, crisscrossing the oil pan back-and-forth for pickup at the front of the pan.

Original oil reservoirs that locate below the transmission had minimal baffling. The result was that some oil did all the work whereas other portions of the oil simply languished in the tank. More complex baffling introduced some years ago forced oil to distribute heat more evenly for added efficient dissipation.

The oil now mixes thoroughly in order to assist in heat dissipation and the suspension-removal of dispersed contaminants from engine wear and combustion blowby.

## The Exhaust

We have all been complaining since 1999 with the introduction of the Twin Cam about sometimes-insufferable heat emanating from the engine and the exhaust under our right and left thighs. The rear exhaust-downpipe-header afflicts the rider from the right side while the crossover pipe

extends out to distress the rider from a left side vantage point.

The 2009 touring bikes address this burning issue from the exhaust through superlative multifunctional design. First we have a performance increase via a collector accepting both front and a forward swept rear pipe that also mercifully takes the exhaust header away from the rider's right thigh. The rear header pipe extends forward in front of the transmission and down to the frame rail to begin extending back. The rear crossover pipe leaves after the collector not up and over the transmission under the seat and over the top rear of the primary system but underneath the bike to connect with the left side rear muffler. The right pipe extends back from the collector to connect with the right side rear muffler. The design relieves some heat from the passenger making his or her ride more enjoyable as a result. There are fewer joints between pipes to seal and clamp. The 3-joints, five if including the header to head connections, connect each

muffler plus the rear crossover pipe to the collector.

A fresh exhaust port gasket (65324-83B), still bases on the initial Evolution design in 1984. Air leaks sucking cooler air past exhaust joints cause the exhaust popping so common with deceleration. New Torca clamping assists in tighter connections and popping should reduce also with fewer tighter fitting joints.

The front exhaust shield is now one-piece completely covering any pipe bluing from the rider's eye.

The only criticism I might have that will be born one way or the other over time is that the crossover pipe might have exposure to damage as it is unprotected by frame members.

California bikes have the catalytic converter inside the

collector. Those bikes with the catalytic converter will now be able to meet emissions dictates with some aftermarket mufflers since most of the pollution control now occurs before the mufflers. I predict the rest of the world will also soon have a catalytic converter as EPA dictates tighten once again.

Rocker box shields

The Ultra Classic, Police, and CVO Ultras also have rocker box shields on either side of the rear rocker box in front of the seat to deflect engine heat for rider comfort. These mid-frame air deflectors are installed during the pre-delivery inspection (PDI) on Ultra Classics,

## TC96 Engine

Flywheels

The endplay specification of the flywheel side play inside the engine casings increases to .003” to .013” (.076mm to .37mm), from .003” to .010” (.076mm to .25mm). This is a significant change although it may seem small.

New retrofit flywheels will accept the now old long compensator hold down bolt or a new short bolt. Early 2009 flywheels will come with the longer bolt as the Factory uses them up. The compensator assembly absorbs engine shock as torque transfers out the sprocket shaft and down the powertrain. Sometimes, this compensator has been bottoming out. There have been temporary fixes by adding an extra spring/shim.

The 2009 compensator is now 7% stronger. Some hope, this move will control power pulses that made the IDS rubber mounted rear pulley necessary to dampen Cruise Drive gear

noise. We shall see.

## TC103 Engine

The TC103 engine powers the 2009 FLHTCUTG Tri Glide trike that I will discuss in detail further on in this chapter. Any changes I discuss in the previous section TC96 Engine is applicable to the TC103 also. The TC103 gains its 7-extra cubic inches from re-boring the TC96 3 ¾" diameter cylinders out the 3 ⅞".

## CVO 110 Engine

My, my the lower rocker box (17573-99A) is now relieved for valve spring clearance, eliminating inefficiencies and noises due to the valve spring retainer touching the rocker box. The CVO 110 receives this change as of December 17, 2007. TC96 and TC103 engine chrome rocker-boxes receive

this change as of March 7, 2008. A new rocker box gasket (16719-99B), matches the rocker box. Be careful not to use the traditional gasket with the new rocker box. The rocker box will retrofit with the correct gasket.

I hope that cylinder base oil-weepage and leakage will eliminate as the old black cylinder base round O-ring replaces with a new orange quad ring. As well, a smaller orange cylinder base drain dowel-pin quad-ring replaces its predecessor black round O-ring. The kit containing these square quad-rings, two for the base and two for the dowel sealing is OEM part number 11907. There is also a center case attachment bolt O-ring to quell leaking from this area.

Please refer to dealer only TT370, May 13, 2008 and TT372, May 22, 2008 for CVO 110 Twin Cam service updates. TT is the acronym for Tech Tip. Mercifully, there is a new

head gasket replacing the excellent flat-surface metal MLS head gasket. The new graphite head gasket (16801-07B) retains the same service procedures, torque sequences and torque specifications. In my opinion, this is a Band-Aid approach but a necessary one, nonetheless, in controlling head gasket leakage from the rear head oil-return dowel-pin.

The automatic compression release (ACR), solenoid (28861-07A), began production as of January 28, 2008. It revises for both the TC103 and the CVO 110. The latest ACR replaces part number 28861-07 and has a new wire boss shape, which requires a new installation tool, part number HD-48498A.

There is a fresh cylinder head for the CVO 110, which has valve guide changes allowing cam lifts up to .596”. The front cylinder head with a new functional ACR is part number

17251-07B indicating this is the third head in use in three years. The rear cylinder head, again with the new functional ACR is part number 17252-07A indicating it is only the second head in use for the current 3-year production of the CVO 110. This is odd, to me anyway, since it is the rear head that has been having the heat-related problems. The Factory makes no note of different casting materials to withstand the excessive heat plaguing the rear-shrouded cylinder head that sometimes warps the head gasket surface causing the old MLS gaskets to blow outwards from the oil-return drain dowel-pin. The new valve guides allow the extra cam lift capability. The tolerance to remember is .080" clearance beyond the valve lift measuring between the top of the valve guide seal fitment over the valve guide and the underside of the top valve spring retainer. The new shorter intake valve guide identifies by a machine groove on the intake-head-port end of the guide. Both the shorter intake and longer exhaust guides also have retention clips. The lower valve collar fits under the retention clip. It is normal to have up to .070" tolerance-play between

the collar and the retention clip. There are no oversize valve guides for this head. Unbelievably, new head replacement is necessary when the guides wear beyond tolerance. I am sure the aftermarket will fill this void.

The old valve guides taper on their port ends. This assists in airflow. The new valve guides do not taper and square on their port ends…one step forward, one step back.

Compression

Heat is the enemy of the CVO 110 as it causes cylinder liner flexing and warping of the cylinder head gasket surface, which cause the rear head gasket to blow outwards from the oil return dowel pin. As I have written many times, compression produces heat because increasing compression creates pressure. Heat and pressure is the same thing.

Expanding heat creates pressure and pressure creates heat.

Twin Cam Compression Ratios

*The exception is the 2009 whose compression reduces to 9.2:1 from 9.3:1.

In a bid to further control CVO 110 heat issues, the Factory reduces compression from 9.3:1 to 9.15:1 on the 2009 FXDSE, and the H-D reduces compression on the 2009 from 9.3:1 to 9.2:1.

On a normal engine like the TC96, which has a 9.2:1 compression, this is not of major concern. However, on a high performance engine like the CVO 110, lowering compression

will certainly assist with heat issues but it is counterproductive to producing torque and horsepower. Any engine, I would build in the 110 cubic inch realm would certainly have a minimum of 10.5:1 compression to develop the necessary oomph one would expect from an engine this big.

Of course, dual 10-core turbulated oil coolers, a larger oil reservoir, ignition and fuel modifications and other changes need to go hand-in-hand along with metallurgical and design changes.

The real problem, of course is that it is impossible to build fast big-inch, air-cooled H-D engines and stay within EPA and CARB guidelines. Harley-Davidson continues the increasingly difficult task of satisfying traditional customer wants with modern emissions dictates.

Harley-Davidson is pretty good with warranty claims and tries to resolve most legitimate customer issues although they do have their moments. In my opinion, many times the Factory will not admit to some problems but work hard in the background until there is retrofit fixes that sometimes H-D refers to as an enhancement. This delay can cause much rider frustration especially when the H-D owner does not know there is an upgrade in the works. Usually, the design of the enhancement has the purpose of extending engine longevity and dependability.

The problems with the first two years of Screamin' Eagle CVO 110 engines is a case in point. Leaking rear head gaskets and warped rear head gasket surfaces are impossible to ignore. Again, these are symptoms of the causal factor, heat

that is mandated by the EPA to burn more harmful emissions. However, the symptom of the shifting cylinder liner is ignored with this particular H-D fix but if the head gasket surface of the rear head will not warp, then it really ceases to become an overt symptomatic problem.

The July 28, 2008 Service Bulletin M-1225; 2007/2008 CVO 110 C.I. DISPLACEMENT ENGINE UPDATE PRODUCT PROGRAM 0905 seeks to upgrade 2007 and 2008 models built before April 9, 2008 at no cost to the customer. Harley-Davidson will have sent a letter to all registered owners advising of this. This free upgrade is for two years after the date of this Service Bulletin, which means it is good until July 27, 2010.

If you are unclear whether your CVO 110 falls under this update, consult any H-D dealer with your VIN (serial

number). The dealer can access the service tab on UltraComm on www.h-dnet.com, which is a secure Harley-Davidson website to advise you if your bike needs this upgrade. I don't think there will be much customer confusion. If your CVO 110 has at least 10,000 miles on it and there is no rear head gasket oil leak then you are most likely not affected. The upgrade kit basically replaces the old heads with updated heads and new ACR automatic compression releases along with new hopefully improved gasketing and the top center motor case bolt. Many ACR's on 2007 and 2008 CVO 110's are defective and may release compression while the engine is running or not work when supposed to ease starting.

There is a different full upgrade (repair) kit for 2007 and 2008 CVO 110's. The 2007 CVO 110 upgrade kit is OEM part number 94676 for the 2007 and the 2008 kit is OEM 94675 for the 2008 models built before April 9, 2008. This is pretty straightforward until the reader gets into the next partial upgrade covering other Screamin' Eagle models. I had to read

everything over about three times before understanding, so please be patient.

There is a separate fix for both 2007 and 2008 specific date Screamin' Eagle CVO 110 models; 2007 FXDSE, FXDTSSE, and the 2008 models built before May 14, 2008 and 2008 models built between April 9, 2008 and May 13, 2008. This upgrade kit as outlined in the July 28, 2008 Service Bulletin M-1226; 2007/2008 CVO 110 C.I. DISPLACEMENT ENGINE UPDATE PRODUCT PROGRAM 0906 does not include new I cannot understand this as in my experience these models also have head failures. The 94677 kits include some specific gaskets for either year but not both so there will be extra gaskets at the end of the enhancement procedure.

2007-2008 Screamin' Eagle CVO 110 Upgrade Enhancements

1 This kit covering both 2007 and 2008 specific date S.E. CVO 110 models (2007 FXDSE, FXDTSSE, + 2008 models built before May 14, 2008 and 2008 models built between April 9, 2008 and May 13, 2008) does not include new heads, which I cannot understand as in my experience they also have head failures. The 94677 kits include some specific gaskets for either year but not both so there will be extra gaskets at the end of the upgrade procedure.

2 I don't get why the problematic rear head is generation denoted by the "A" suffix on the part number while the normally okay front head is generation denoted by a "B" suffix. I would think it would be the other way around but what do I know?

3 On the 2007/2008, 2007 and 2008 upgrades, the top center engine case bolt replaces with the supplied 1090A

screw, hex flange. Specification torque is 15 to 19 ft-lbs (20.3 to 25.8 Nm). The 1090A screw (bolt) replaces the OEM 1090, 5/16”-18 x 3 ¾”screw, hex flange (2007 to 2008), which replaces the predecessor OEM 895, 5/16”-18 x 3” screw, hex flange in use since the advent of the Twin Cam in 1999.

4 The “A” suffix indicate a generation head. Therefore a modification to the initial 2008 head has been done. This can be as simple as a paint color or as complicated as a major redesign.

The clamp OEM 10038A is a gas line clamp for the equalizer line at the front of the gas tank, which must be removed in the disassembly process.

1. Gasket OEM 29313-95 is an HDI (H-D International) part that fits between the air filter and the air filter backplate

assembly (backing plate) for the 2007 update. This gasket saw first use on all 1995 Big Twin Evolutions.

2. Therefore, gasket OEM 29645-08 is also an air filter gasket seeing first use in 2008. It is for the 2008 CVO 110 update.

3. Gasket OEM 29583-01A is the 2007 update gasket for the air cleaner backing plate to the EFI induction model that saw first use as generation OEM 29583-01 in 2001.

4. Gasket OEM 29059-88A is the gasket for the air cleaner backing plate.

5. Gasket OEM 29593-05A is another air cleaner gasket. The kits must have gaskets to cover numerous models and applications.

6. Gasket OEM 29718-08 is the 2008 update gasket for the induction module to the air cleaner back plate.

7. The two OEM 11292 O-rings are for the 2007 update to seal the breather bolts that screw into each head from the inside of the air cleaner. These are not provided or needed for the 2008 update, as there are different breather bolts and sealing system.

CVO 110 Cylinder Condition

There is a specific testing procedure to determine if deglazing cylinders and new ring installation is necessary by Motor Company standards.

What do I do? I deglaze all removed cylinders and I use new rings all the time in every instance. Fresh bored cylinder

bores require that the top compression ring be moly (molybdenum) while deglazed cylinders should use cast top compression rings, which are softer.

H-D specifies a very comprehensive leakdown test in Service Bulletin M-1226 that determines if compression is seeping by the rings. Power is all about compression. This 3-phase leakdown test is really three tests in one whereby leakage is tested with the piston rings at TDC (top dead center), half way down the cylinder bore and at BDC (bottom dead center). Do the in-between mid-stroke leakdown test with the bike in sixth gear and the rear brake activated. Leakage of 10% or more in any of the three positions requires deglazing and new rings. As I have said, I personally deglaze and replace rings in all cases.

## The Transmission

There is a new transmission top vent line (62372-09) as well as new transmission locking rings (2), with serrations on one face. The smooth side faces the gear.

The Softail and Dyna models retain the same internal gear ratios in use since the introduction of the 6-speed Cruise Drive transmission.

Touring models have new gear ratios that result in better acceleration for the heavier bikes.

There is also a new 68-tooth rear pulley. It requires a 140-tooth rear belt. This changes secondary and final gear ratios. Please see the charts below.

USA Sprocket Sizes: 1999-Present

The reason that 2006 Dynas and 2007 and 2008 Big Twins have differing gear ratios is because of a 2-tooth/nub reduction in the 2007 rear wheel pulley. The 2009 models go up 2-nubs on the rear pulley and have the same. This will improve acceleration for the 2009 models.

## Sprocket/pulley Sizes

## Primary: Secondary and Final Drive Ratios

The reason that 2006 Dynas and 2007/2008 Big Twins have differing gear ratios is because of a 2-tooth/nub reduction in the 2007/2008-rear wheel pulley. The 2009 models go up 2-nubs on the rear pulley and have the same primary, secondary and final drive ratio as the 2006 Dynas.

## USA Overall Gear Ratios

H-DI (H-D International) models generally have taller

gearing for better highway touring because of increased speed limits. International models usually have smaller rear wheel pulleys than those used in USA

Internal Transmission Ratios for each gear

Performance is all about increased horsepower and torque. The third component, drive ratios, delivers this power in a way specific to the rider needs and wants.

S&S uses a 34-tooth Transmission pulley with 70-tooth rear pulley.

Overall gear ratios show the number of engine revolutions required to turn the rear wheel one 360-degree turn. For

example in gear all 2007-2008 Twin Cams and 2009 FXST and Dyna models take 9.31 engine revolutions to rotate the rear wheel once while a 2009 Touring TC96 takes 9.59 engine revolutions to rotate the rear wheel once. A 1999 Big Twin takes 10.81 engine revolutions to obtain the same goal in first gear. Overall gear ratios take into account the internal transmission ratios with the primary and secondary gear ratios while the final drive ratio is a simple multiplication of the primary ratio times the secondary ratio.

## Cruise Drive 6-speed Transmission Gear Ratios

## 2009 140 -tooth Rear Belt Tension Specifications

## Miscellaneous Touring Changes

A new exhaust Torca clamp (65296-95A) has an assembly friction reducing coating for ease of installation that doubles as a secondary locking mechanism after tightening. Tightening torque is 38 to 43 ft-lbs (51.5 to 58.3 Nm).

The new shorter AM/FM antenna retrofits back to 1998.

The CB will require removal of the loading coil in the Tour Pak carrier. The CB antenna is marked for identification.

The new Tour Pak carrier support structure increases the rated payload by 5 lbs. It still uses the current Tour Pak. The rear lighting uses the current tail lamp and turn signals with revised light bars, new license plate mounting and the rear tip light is now LED The saddlebags move back .5”. There are new support structures and guards. The rated payload increase for each bag is 5 lbs. There is a stator plug material change to correct the former plug’s tendency to pull through the left engine casing.

Fork oil for Road King models is 11.0 ounces (125mm). Other touring models take 10.7 ounces (122mm).

The fork-stem nut now has no lock washer and the nut torque specification is 70 ft-lbs to 80 ft-lbs (94.9Nm to

108.5Nm). The steering head adjustment now stops on the or swing. There is also a new fork triple tree.

## 2009 Tire and Wheel Information

### New Model and Year Specific Tire Tread

The 2009 FLHRC Road King Classic as well as the FXDF Dyna Fat Bob, and the FXDFSE Screamin' Eagle Dyna Fat Bob use a new front tire tread on the 130/90 B16 tire.

Front PSI tire-fill requirement is 36 lbs on all touring models.

Rear touring models tire-fill requirement is 40 lbs for solo rider and 2-up.

FXDF, FXDFSE rear tire fill-requirement is 36 lbs for solo rider and 40 lbs for 2-up.

This front tire tread will not retrofit correctly to earlier models because it will not match older rear tire treads. Furthermore, do not attempt to put older model tires on the 2009 FLHRC. There are new Dunlop tires on all touring models. They are model specific. Be very careful, checking with the manufacturer for matching tire availability if retrofitting forward or back in years.

All other touring models excepting the FLHRC (16” wide-white-wall tire) now use a new cast 17” (130/80-17) front wheel.

Touring Model Rear Wheel and Associated Parts

The rear tire is now a whopping 180mm, with a lower 65 profile that fits on a new 28-spoke 5.00” wide 16” rear wheel. Traditional H-D rear tires are a narrower 130mm with a higher 90mm profile. The new rear tires are multi-tread. This means the center part that has use when traveling straight is a harder compound for longer lasting wear characteristics while the sidewalls are softer and stickier gripping for turning purposes.

Furthermore, there is a new rear wheel lacing procedure that will catch mechanics offside without consulting the H-D manual.

The wheelbase increases by .5-inches.

Of course, this wider fat rear wheel necessitates a fatter rear fender that closes gaps and is sleeker.

There are new tire direction specifications for the FXCW and FXCWC. Consult your manual.

The tread design on the 2009 FXCW and FXCWC reverses from Any mechanic will inadvertently put this tire on backwards through many years experience in observing tire tread patterns. This will lead to many incorrect tire installations in the future for those that do not study new technologies and procedures.

There are many new and sometimes-subtle changes that the rider and/or mechanic should not take for granted.

Rear Pulley Changes

The 2009 compensator gear remains at 34-teeth, as does the clutch sprocket at 46-teeth. The rear drive transmission-pulley also remains at 32-nubs but the rear pulley increases by 2-nubs from 66-nubs to 68-nubs resulting in better acceleration. The increase in pulley size requires a new 140-tooth length belt. The primary, secondary, and final drive ratios are now the same as in used on the introductory 2006 Dyna models. Please see the gear ratio charts that I have drawn up in this chapter.

The IDS (Isolated Drive System), rubber mount rear drive pulley compensator is now part of the rear wheel hub and separate from the rear drive belt pulley. The rubbers in the compensator absorb power pulses from the engine traveling down the powertrain. Before developing the IDS, pulses were returning back up from the former rigid mount rear pulley

causing the Cruise Drive spur gears to chatter. The pulley is now just a pulley with no dampening function.

2009 Front Tire Pressure Chart

2009 Rear Tire Pressure Chart

Do not exceed the maximum cold tire inflation pressures that the rider will find molded or branded into the sidewalls of all tires. Front tire pressures are always a few pounds lighter than the recommendations for the rear, as weight distribution is less. These inflation figures are a base generalization only for average size riders and passengers. Conditions will modify the pressure up or down. For example,

high-speed riding or very high temperatures may require a moderate increase in pressure. Follow tire manufacturer recommendations. Sizing may indicate in inches, metric, or by code alphanumeric lettering. For example, if tire width is 5.00-inches, this translates into 130mm in metric. The alphanumeric letter coding is MT for 5.00”, which is the same as 130. Tire width is always the first part of the sizing nomenclature. Tire height or aspect ratio to width is the second. Aspect Ratio equals the height/width x 100. Tire height usually expresses in metric millimeters (mm). Therefore, if the stock height or aspect ratio Harley tires designates as 90, this translates into 3:25 inches. A lower profile, (height or aspect ratio), of 80 is smaller with 70 smaller again and so on.

## Tires: Inch, Metric, and Alpha Designations

PR = ply rating. The letter B sometimes precedes the last figure, which identifies the rim diameter or comes after it. B indicates a bias ply tire. The letter R indicates a radial tire.

## Motorcycle Speed Ratings

The acronym kph stands for kilometers per hour. Older H-D tires had an H rating, which means the tire can run at sustained speeds of 210 kph or 130 mph. A V rating indicates sustained speed capability at 240 kph or 149 mph. Therefore,

wide safety margins exist for H-D tires since they can never achieve the speed ratings.

## Common Harley-Davidson Tire Sizing

Many charts from different sources and manufacturers vary on sizing and designations. Use this chart as a general guide only as no chart is completely accurate. There are some new rear tires for fitment on the FLHTCUTG Tri Glide trike whose wheels are matching cast aluminum 7-spoke units. The front is a 16” using a traditional MT90B16 tire. The dual rear 15” automotive style wheels fit with P205/65R15 radial tires.

### Softails

There is a new common to-all-model electronic control module (ECM). The ECM has improved idle air control (IAC) software as a running change in January 2008.

The FLSTN has a new seat with collapsible sides. Collapsible pockets in the filler material relieve pressure to the rider inner thigh contact area.

New parts include, rear shocks and bumpers, exhaust Torca clamps and mounts, jiffy stand (sidestand) sensor, FLSTC front brake line, FLSTC handlebars, risers and top clamp; new front fender on FXST, FXSTC, and FXSTCB models plus other minor changes. This listing is not comprehensive with minor styling and functional changes. Consult your dealer.

Dynas

FXD, FXDC, and FXDL models have no new technical content. There are also no new service procedures for the Dyna family. This may sound strange but the Dyna family has gone through many dramatic technical and model changes in the last three years.

All models have new silver-faced gauges. There is also a range-readout as part of the fuel tank sender indicator on the speedometer. Life is getting cushy on a Harley.

The FXDF Fat Bob and the CVO Fat Bob have new rear shocks.

FXDB

The FXDB Street Bob has a LED retro-style tail light with integrated LED stop-turn taillight signals on a novel bobbed

rear fender. There are no traditional chrome fender support covers.

There is a lower 25.8” solo seat.

Handlebars wire internally.

There are currently cool black wrinkled belt guards, battery cover and fuel tank console with gloss black wheel rims.

The powertrain is black and silver without highlighted finning. Japan Harley’s have highlighted finning.

Lastly there are mid-mount foot controls for riding stability, a custom air cleaner insert plus chopped straight-cut mufflers.

# The FLHTCUTG Tri Glide Trike

The new design of the FLHTCUTG bases around an entirely new frame.

## The Acronyms are out of Control

Boy-oh-boy, H-D is really stretching the limits of an acronym. It is okay for the old Harley people like me that can dissect the letters and break them down. FLHT is a rubber mounted base model-touring bike. C is the designation for an upgrade Classic, which includes a stereo and tourpak while the U is for the top model Ultra.

Whoops, used to be but not quite so anymore because of

the FLHTUSE, which is a Screamin' Eagle Ultra. Mercifully, the Factory dropped the "I" as in FLHTCUI, which indicates that the bike/trike is fuel injected to separate them from carbureted models that do not exist after 2006. The TG stands for the Ultra Trike, more correctly known as the Tri Glide. Yep, the acronyms are getting too long for the ordinary Harley rider who must memorize all this stuff just so he or she can sound educated about their prestigious ride.

The Trike

The new chassis bases on a Lehman Trike design. Lehman's is one of the major trike players for Japanese and Harley two wheel motorcycles. I have installed Lehman kits for over ten years now. They are very safe and base on simple but effective technology. However, in my opinion they are not sophisticated, relative to some competitor kits.

However, the Tri Glide is not a kit bike but a completely new motorcycle that brings excellent H-D design, fitment and paint coordination. They will serve the Harley rider well once steering improves a little more.

Steering

H-D has already made some of the prerequisite changes to improve steering geometry by reducing effort by a claimed 25%. The front forks extend 1.775” over the two-wheel touring forks and rake increases from to In addition, there is an essential steering stabilizer dampener to assist in steering. These changes are necessary to eliminate steering shudder at some rpms from road irregularities and steering. They do not eliminate yet but I have full faith in the H-D engineers to work their magic. Still, riding this introductory model is a lot of work until muscles adjust.

The Flathead 45 Servi-Car

Harley-Davidson is not new to trike building. The Motor Company discontinued the 45c.i (750cc) flathead GE series Servi-Car production in 1973 after long and illustrious service. Any readers that have ridden a four-five, which is Harley vernacular for this size engine, will know that they are slow and not capable of modern highway speeds.

The Tri Glide is a fast Harley. It is capable of excellent acceleration with the Cruise Drive 6-speed transmission and effortless high touring speeds thanks to the TC103 engine that is 2.3 times bigger than the four-five but delivers much more than 2.3 times the performance.

The four-fives used mostly plumbing-faucet Schebler and Linkert carburetors whereas the TC103 engine utilizes H-D's

sophisticated engine management system with and ESPFI electronic sequential port fuel injection.

## The Rear End

The new Tri Glide rear-axle assembly uses an aluminum center section with steel axle tubes. Final drive is with H-D's new carbon fiber belt drive. The rear suspension features the new 2009 dual air-adjustable rear shock absorbers.

## Riding a Trike

A handlebar-mounted reverse module engages an optional electric reverse that integrates with the rear differential assembly. I find it surprising that this is a option. Everyone on a trike wants a reverse although I can say from experience that a trike is easy to muscle around unless suffering from a

disability. However, pushing one backwards up an incline is not easy at all, so an experienced triker is always conscious of parking in favorable exit conditions.

Traditionally, reverse systems for H-D aftermarket kit trikes have not been very good, which does not necessarily reflect on the functionality of the new optional H-D unit. Braking consists of the excellent stock touring dual Brembo front disc brakes and a Hayes dual-disc rear brake system with a lever actuating, integrated parking brake.

Trike riding is freaky at first when used to two-wheels. The rider may lean but it has no effect on steering anymore. The procedure becomes pulling on the handlebar end in the direction of turn and pushing on the opposite handlebar end. This must continue for the full radius of the turn as the trike will naturally begin to straighten out if pulling and pushing pressure lessens or stops.

Donny’s Blasts from the Riding a Trike the Wrong Way

I had forgotten about this one until writing on how to steer a trike. The embarrassment lasted for many years.

Around forty years ago, the boys and I were on a run to a rock festival. We set up camp near a small lake and proceeded to imbibe. No one considered responsible drinking back then as plumes of marijuana smoke wafted through the hot summer air.

Acid-rock bands played continuously in the background. Speaking of acid, I avoided drinking from open bottles; especially wine because others sometimes thought my mind needed hallucinogenic expansion.

My buddy was riding a chopped 45-trike with a raked and overly long springer front end. The spool front wheel-hub was brakeless in concert with the non-functional styling of the hippie era. Long upswept fishtail exhausts snaked up each side of the trike's rear beer box high into the air. The wild flame metallic-speck paint job made this trike a popular hangout location.

My pal told me to take her for a spin. I had never ridden a trike but that was not a concern. Macho testosterone laced with booze made the fateful ride's trajectory a foregone conclusion.

I expertly kick-started the trike, activated the suicide clutch and hand-banger shifter whilst turning the deadman throttle. I could do all this by rote even though I had much to drink.

Off I went towards a Johnny-on-the-Spot before the lakeside. No problem as I leaned to the right to circumnavigate the outhouse and make a wide sweeping turn away from the lake to go for a blast. The trike was not turning! I leaned further to no avail. The trike was gaining speed as my friends yelled steering instructions. Turning the front wheel would certainly result in the bike flipping. I had a two-wheel mindset on a three-wheel trike that could not lean.

I moved the deadman-throttle back to an idle position as I careened off the outhouse and finally settled with the front wheel in the lake. At least, I knew to turn off the throttle that aptly earned its name because it stays in whatever position the rider turns it to.

The whole rest of the weekend, I kept a brave face as I

miserably listened to the countless retelling of my misadventure and the endless jokes about my riding skills or lack thereof.

## Trike Wheels, Options, and Bodywork

The wheels are matching cast aluminum 7-spoke units. The front is a 16” using a traditional MT90B16 tire. The dual rear 15” automotive style wheels fit with P205/65R15 radial tires.

The Tri Glide, of course comes with all the Ultra staples and options like cruise control, GPS and a Harmon Kardon stereo. The new Tri Glide rear end body components integrate nicely with the existing Ultra bodywork including the Tour Pak luggage compartment and the passenger seating area. I have built many trikes with kits and there is some skill to matching fiberglass rear ends to stock sheet metal. The

molded-composite rear trunk provides 4.5 cubic feet (127 liters), of storage space and accesses through a rear hatch. The trunk and Tour Pak have 6.56 cubic feet (186 liters), combined capacity.

The current Tri Glide Ultra Classic FLHTCUTG comes in three colors, vivid black, dark blue pearl and red-hot sunglo.

## Researching Service Recalls for all Years

The United States National Highway Traffic Safety Administration (NHTSA) has an office called the Office of Defects Investigation (ODI). Among other things, this department takes care of recalls. The reader can learn whether his own vehicle has any recalls by visiting http://www-odi.nhsta.dot.gov/cars/problems/recalls/recallsearch.cfm.

The odd times I searched the web site I ran into technical difficulties with my browser. The reader can also go to www.nhsta.gov and click on Recalls.

As an example, I searched 2007 Harley-Davidson FXD at the time of writing in April 2007 and found that 614 certain

Dynas and Sportsters were built with an incorrect front tire. The site gives the Harley recall number as No. 0129 with a corresponding NHTSA campaign ID number: 07V025000.

Harley-Davidson has also announced two voluntary service recalls that affect more than 28000 bikes from model year 2007.

The first is Harley-Davidson No. 0127 with a corresponding NHTSA campaign ID number 06V474000 and is in my opinion very serious. On 10,397, FXDWG 2007 Dyna Wide Glides, the hole in the handlebars where the handlebar wiring exits may crack.

One of my fears has always been that the handlebars might break or separate from the front end.

To quote the NHTSA site, “Dealers will replace the handlebar with one with a greater wall thickness and with a wire exit hole that has a shape that reduces stress concentration.” 1

The second voluntary recall, No. 0128, concerns eight models of the 2007 Sportsters. On these bikes, the rear skirt of the front fender can catch onto the top of the voltage regulator body hindering steering. The dealer will relocate the voltage regulator.

Recalls may not affect every model produced so it is important to check if your particular bike is on the recall list at Harley-Davidson 1-414-342-4680. You may also contact The National Highway Traffic Safety Administrations vehicle safety hotline at 1-888-327-4236 (TTY 1-800-424-9153) or go to www.safercar.gov.

If the reader researches the site, it will become quickly apparent that there have been many recalls over the years that the riding public may never hear about. Of course, affected owners are sent letters, etc., but people move or may not always get their mail.

The site is very cool. It goes back to 1949. I had a look at 1970 for the King of the Highway, the FLH Electraglide, and found that 12,963 units possibly had improperly tightened wheel-bearing locknuts. I thought that number would be around 35% of total production, which is very significant.

The Harley-Davison Data Book says that not that many FLHs were produced in 1970, so the recall may have affected other models also. In fact, the book says that there were about 17,000 Harley's, including the GE 45 cubic-inch Flatheads and Sportsters. 2 In any case, one can get lost in this stuff or at

least I can.

I did a search for 2009 recalls in August 2008. There were none as of August 11, 2008.

A check for 2008 yielded a recall for the 2008 FXDWG, the Dyna Wide Glide. Service Bulletin number M-1215B (NHTSA: 10025364), as of January 29, 2008 advises that there are updated parts and service procedures to stop the loss of clamping load for the top steering stem nut.

This is a great site with tons of valuable and interesting information about your Harley-Davidson.

1 The National Highway Traffic Safety Administration (NHTSA), Office of Defects Investigation (ODI); http://www-odi.nhtsa.gov/cars/problems/recalls/results.cfm campaign ID numbers 06V474000. Harley-Davidson. Year 2007, Model:

FXDWG. Service recall 0127, Dec. 12, 2006.

2 Connor, Rick. Harley-Davidson Data Osceola, Wisconsin:

Motor Books International, 1996. Page 115.

# Acronyms

# = number

Ω = ohms

02 = oxygen as in oxygen sensor

A = Alpha (Twin Cam A engines are rubber mounted)

A = amps or amperes (electricity)

ABDC = after bottom dead center

ABS = anti-locking braking system

AC = alternating current

ACC = accessory

ACR = automatic compression release

AFR = air fuel ratio

AGM = absorbed glass mat (battery technology)

AHDRA = All Harley-Davidson Drag Racing

AKS = anti knock sensor

AMP = amperes

AMPS = amperes (electricity)

aps = average piston speed

AR = anti-reversion also known as tork or torque valves

ARP = Automotive Racing Products (superior hardware fasteners)

ATDC = after top dead center

ATS = air temperature sensor

AWG = American wire gauge

B = Beta (Twin Cam B counterbalanced engines)

B+ = battery voltage

BAS = bank angle sensor

BBDC = before top dead center

BDC = bottom dead center

bhp = brake horsepower also known as engine horsepower

BTDC = before top dead center

BTU = British Thermal Unit (energy to raise the temp. of 1 pound of water 1 degree

C = Celsius

c.i = cubic inches

C0 = Celsius temperature degrees

CA = California

CAL = calibration

CARB = California Air Resources Board

cc = cubic centimeters

CCA = cold cramping amps

CDI = capacitor discharge ignition

cfm = cubic feet per minute

cfm per c.i = the ratio of cubic-feet-per-minute air per cubic inch engine displacement

CKP = crankshaft position sensor

CLEFI = closed loop EFI

cm = centimeter

CMP = camshaft position sensor, also known as CPK

CNC = computer numerical control, precise and

computerized automatic machining

CO = carbon monoxide

CO2 = carbon dioxide

CPK = camshaft position sensor, also known as CMP

CR = compression ratio

CV = constant velocity

CVO = Custom Vehicle Operation

dB-A = decibel exhaust noise rating

dbhp = drawbar horsepower (used to measure locomotive and farm implement power)

DC = direct current

DD5 = BAKER 5-speed transmission, 1:1 final gear ratio

DD6 = BAKER 6-speed transmission, 1:1 final gear ratio

DD7 = BAKER 7-speed transmission, 1:1 final gear ratio

DFO = Digital Fuel Optimizer also known as TFI

DIN = Deutsches Institut für Normung. Correction factors for dyno measuring

DLC = data link connector

DOM = domestic

DSPD = digital signal processing dynamometer

DTC = diagnostic trouble code

DVOM = digital volt ohm meter

EC = exhaust centerline angle

ECM = electronic control module (same as ECU or ICM)

ECT = engine coolant temperature

ECU = electronic control unit (same as ECM or ICM)

ED = exhaust duration

EEC = European Economic Community, the former name of the European Community correction standards again bring horsepower figures slightly higher than the SAE.

EEPROM = electrically erasable programmable read only memory

EI = employee involvement

EITMS = engine idle temperature management system (idle temperature cool down)

EMS = engine management system

EPA = Environmental Protection Agency

ESPFI = electronic sequential port fuel injection

ET = elapsed time

ET = engine temperature (usually describes a sensor)

ETC = electric throttle control also known as fly-by-wire

EVAP = Evaporative emissions control system

F = Fahrenheit

F0 = Fahrenheit temperature degrees

F6F = BAKER 6-speed fix gear noise)

F6R = BAKER 6-speed reverse

FL = police model Panhead or Shovelhead

fl oz = fluid ounce

FLH = citizen model Panhead or Shovelhead

= FLHT Classic Ultraglide Screamin' Eagle

FLHTCUTG = Tri Glide (Ultra Trike)

FMS = fuel management system

fpm = feet per minute

ft/s = feet per second

FX = kick start Shovelhead Big Twin sport model 1972 up

FXE = electric start Shovelhead Big Twin sport model 1973 up

FXR = rubbermounted, electric start Shovelhead or Evolution sport model 1982 -1994

g = gram

gal = gallon

GAWR = gross axle rate rating

GDP = gross domestic product

GND = ground

gpm = gallons per minute

HC = hydrocarbons

H-D = Harley-Davidson

HDI = Harley-Davidson International

HDMOCO = Harley-Davidson Motor Company

HDMoco = Harley-Davidson Motor Company

HDOA = H-D Owners Association

H-DSSS = Harley-Davidson smart security system

HFSM = hands free security module

Hg = Mercury

HOG = Harley Owners Group

HP = horsepower

hp/kg = horsepower per kilogram; see also watts/hp, kw/hp and hp/lb (power-to-weight ratio)

hp/lb = horsepower per pound; see also hp/kg, kw/hp and watts/hp (power-to-weight ratio)

HTCC = High Torque Combustion Chamber

IAC = idle air control

IAT = intake air temperature (usually describes a sensor)

IC = intake centerline angle

ICM = ignition control module (same as ECM or ECU)

ID = inside diameter

ID = intake duration

IDS = Isolated Drive System (rubber mount rear drive pulley)

IGN = ignition switch (ign)

Ihp = indicated horsepower

IM = instrument module

in. – inch

INJPW = injector pulse width

in-lbs = inch-pounds

IST = intelligent spark technology (S&S ignition)

ITC = International Trade Commission

J = joule (a joule is the mechanical equivalent of heat).

JIS = Japanese Industrial Standards. Correction factors for dyno measuring standardization in Japan, adjust horsepower figures slightly higher than the SAE.

JIT = just in time inventory

k = kilometer

K = kilometer

k = thousand (1000 meters in a kilometer)

k/hr = kilometers per hour

kg = kilogram (2.2 lbs)

kgf.m/s = kilogram-force meters per second

km = kilometer

km/hr = kilometers per hour

= metric pressure measurement equivalent to psi

kPa = kilopascal

kw/hp = kilowatts per horsepower; see also watts/hp hp/kg and hp/lb; (power-to-weight ratio). 1 horsepower equals .7457 kilowatts

l = liter

L/R = connecting rod length divided by the engine's stroke length

LC = lobe centerline angle

LCA = lobe centerline angle

LCD = liquid crystal display

LED = light emitting diode

LSA = lobe separation angle

m = meter

m/s = meters per second

mA = milliampere

MAN = material as needed

MAP = manifold absolute pressure (usually a MAP sensor measures manifold vacuum)

MCR = mechanical compression release

ml = milliliter

MLS = multi layer system; refers to multi-layered gaskets

MM = Magneti-Marelli electronic fuel injection used from 1995 to circa 2001-2002.

mm = millimeter

Mo = molybdenum

MOCO = Motor Company, a colloquialism for Harley-Davidson

Moco = Motor Company, a colloquialism for Harley-Davidson

mph = miles per hour

mps = mean piston speed

ms = millisecond

MSO = Manufacturer's Statement of Origin

MY = mid-year

N/A = not applicable

NA = not applicable

nhp = nominal horsepower (inaccurate and out-of-date)

Nm = Newton-meter (Sir Isaac Newton), a metric unit of torque

No = number

NOx = nitrogen oxides

O = ohm (electrical)

od = outside diameter

OD6 = BAKER 6-speed overdrive transmission

OD6-Z 0.86 = BAKER 6-speed overdrive transmission; 0.86 final gear ratio

OEM = Original Equipment Manufacturer (stock parts)

OHC = overhead cams (VRod has overhead cams)

OHV = overhead valves (all Big Twins and Sportsters have overhead valves)

OLEFI = open loop EFI

oz = ounce

P = phosphorus

P&A = parts and accessories

P&P = porting and polishing

Pb = lead

PC = Power Commander

PCI = Power Commander I generation)

PCII = Power Commander III generation)

PCIII = Power Commander III generation)

PDA = personal digital assistant (handheld computer)

PHD = H-D technician training at Harley University (professional Harley-Davidson)

Pi = Pi or π is a mathematical constant; 3.14159. It represents the ratio of a circle's area to the square of its radius.

PN = part number (pn)

psi = pounds per square inch

pt = point (as in 12-point or 12pt hardware fastener)

R&D = research and development

RA = roughness average, micro-inch measurement of a surface finish like a gasket surface.

RES = reserve

RFI = radio frequency interference

rpm = revolutions per minute

RSD = right side drive (transmission for big rear tires)

S&S = S&S Cycle, Inc.

SAE = Society of Automotive Engineers (correction factors for dyno measuring standardization in USA/Canada since 1990)

SCFH = cubic feet per hour at standard conditions

SE = Screamin' Eagle

S.E. = Screamin' Eagle

SERT = Screamin' Eagle Race Tuner

SOC = statistical operator control standardization in Germany, adjust horsepower figures slightly higher than the SAE.

STD = same as STP. Correction factors for dyno measuring standardization in USA/Canada before 1990, adjust horsepower figures slightly higher than the SAE.

STP = same as STD. Correction factors for dyno measuring standardization in USA/Canada before 1990, adjust horsepower figures slightly higher than the SAE.

SV = side valve Flathead

TB5 = BAKER 5-speed torque box transmission

TB6 = BAKER 6-speed torque box transmission

TCA = throttle control actuator

TDC = top dead center

TFI = Techlusion Fuel Injection also known as DFO

TGS = twist grip sensor

thp = true horsepower also known as rear wheel (whp) or effective horsepower

TMAP = intake air temperature/manifold absolute pressure

TP = throttle position

TPS = throttle position sensor

TSM = turn signal module

TSSM = turn signal/security module

TT = Tech Tip (H-D vernacular for circulars that it sends out to its service people)

TÜV = European Community standards (similar but different from EPA)

TV = Terminal Velocity (EFI fuel modifier)

US gal/m = U.S. gallons per minute

V = volt

V/P = volume to pressure index (monitors volume of air/ gas relative to compression ratio)

V2 = Evolution

VAC = volts of alternating current

VDC = volts of direct current

VE = volumetric efficiency (usually refers to amount of cylinder fill)

VFI = variable fuel ignition (S&S EFI)

VIN = vehicle identification number. Harley's register by

the frame VIN since 1970 and the engine 1969 and earlier.

VO = valve overlap

VOES = vacuum operated electric switch

VSS = vehicle speed sensor

W = watts (electricity)

watts/hp = watts per horsepower; see also kw/hp, hp/kg and hp/lb; (power-to-weight ratio) 1 horsepower equals 745.7 watts.

WEGO I = Wide-band Exhaust Gas Oxygen: first generation (Daytona Twin Tec tuner)

WEGO II = Wide-band Exhaust Gas Oxygen: second generation (Daytona Twin Tec tuner)

WEGO III = Wide-band Exhaust Gas Oxygen: third generation (Daytona Twin Tec tuner)

whp = wheel horsepower or rear wheel, true (thp), or effective horsepower

WO2 = wideband oxygen sensor

XL5 = BAKER Sportster 5-speed

XL6 = BAKER Sportster 6-speed

$\pi$ = Pi is a mathematical constant; 3.14159. It represents

the ratio of a circle's area to the square of its radius.

# Author Biography

When the Nazis' invaded his parent's homeland, Denmark, during WWII both Donny's parents, émigrés to Canada, joined the Royal Canadian Air Force, becoming commissioned officers by war's end.

Enrolled in an inner city school as a child, wearing European britches and knee socks was not a pleasant experience. After many beatings, Donny began to learn how to fight or rather the need to fight.

Trying to fulfill every immigrant parents dream for the first born, Donny was not allowed to go to technical schools and wound up at Brock and later York University, in Toronto

Canada, studying Town Planning. After three years of post secondary school education, Donny surveyed his class of 300 realizing that only six jobs existed at the time for this occupation across Canada.

Donny decided to pursue Social Work, which he had been doing part time as an outreach youth worker with the YMCA. Working with street gangs in Toronto's toughest housing projects on the streets, at night was always interesting and sometimes dangerous.

During this time, he purchased his second Harley, a 1966 Shovelhead. Donny began to work on his bike. He found this very satisfying, as there is a cause for every effect and vice versa. This was much easier than fixing the problems of wayward youth.

With the early Seventies advent of psychedelic drugs, Donny was thrust into crisis intervention on the streets having to deal with bummers, freak-outs and overdoses and all that went with them. Outreach workers were way ahead of the curve as society tried to grapple with this new counter culture. Donny helped form the first non-counter culture, street-level counseling center in Ontario under the auspices of the Drug Addiction Foundation and the YMCA. His first teaching experiences followed with parents groups, social workers, psychiatrists and a failed attempt with the Metro Toronto Police College. Donny tried to teach young police recruits on how to deal with youth having bad trips on the psychedelics without using force or restraints, which could easily lead to permanent impairment. The only thing the recruits wanted to learn was how to cut Donny's mid-back length ponytail.

Donny had been watching the early biker movies but when Easyriders premiered his course in life became set. Soon after,

he proudly finished his first chopper, a day-glow painted, radically raked, twelve inch extended wide glide with no front fender and ape hanger handlebars.

The same summer while riding, he was center punched by a car in an intersection, breaking his back and leg in over a hundred places. A medical journal documented Donny's miraculous cure after refusing to have his leg amputated at the hip. Doctors felt this necessary because of a crushed femur and a mostly severed main nerve before the advent of microsurgery.

Lying in traction for months without being able to move taught Donny his most prized life skill. The options were going crazy or developing a strong disciplined mind. He developed the mind and became a heavy weight lifter, when able, to develop the body.

On crutches, with a 1959 Panhead, replete with suicide clutch and hand shift, he began his 37-year membership sequentially in two prominent motorcycle clubs.

A few years later, the urge to become entrepreneurial overcame him. He worked out of back alley garages in Toronto's tenderloin before formally starting Heavy Duty Cycles in 1974. In 1977, Donny became a licensed mechanic. Post secondary school physics taken many years before came into play.

Physics allowed Donny to understand and develop hi-performance concepts, which helped immensely in his business.

Those school essays also taught him how to write. Living on the streets as well as in academia in addition to a blue-

collar trade gave Donny interesting perspectives on life.

As his businesses developed, a manufacturing stint in the Orient made Donny realize that he loved to travel, which has become his main passion other than motorcycling.

Donny became a tech writer in 1989. His articles have worldwide readership. This in turn has given Heavy Duty Cycles international stature.

His motorcycles, many hand fabricated, are featured in publications from Great Britain, France, Japan and Russia as well as closer to home in magazines such as Easyriders, VQ., American Iron, Hot Bike and Canadian Biker. Donny was first titled as a “Master Builder” by Supercycle Magazine in the 1980’s and then awarded this status by VQ and Easyriders Magazines but Petersen is the first to acknowledge that no one

person can fabricate the bikes that Heavy Duty Team always aspires to.

It requires a team effort by the industry's best and Donny confers this honor on the Team of which he is only a part.

Donny began to teach every apprentice motorcycle mechanic in Ontario, before they were able to test for their mechanics license. He then became involved with the Federal Government and the Provinces in updating and standardizing these courses for the whole country. Donny followed up by assisting in writing the final computer generated Mechanics Exam utilized across Canada in order to produce standards respected worldwide.

The Canadian Government invited Donny to teach the Havana Harley Riders Club at the Embassy in Cuba as a goodwill gesture to the people in 1997 looking forward to a

post-Castro democracy.

Petersen with committed friends spearheaded many bikers' rights initiatives over the years mostly to no avail. One constitutional challenge to Canada's Supreme Court led to another honor that of being invited to speak to the prestigious Empire Club in Toronto's Royal York Hotel. Fellow speakers over the years included George Bush Sr., Ronald Reagan, Billy Graham, Dalai Lama, and Queen Elizabeth. Donny's invitation distinguished him as one of the few no-name speakers.

An avid scuba diver, Donny's first open water dive was in the South China Sea off the coast of Borneo looking for whale sharks.

Donny's other hobbies include mountain trekking. He has trekked the Inca trail from the center of the Incan empire in

Cuzco in the Andes to Machu Pichu in Peru. Petersen then climbed Africa's highest mountain, Kilimanjaro.

Donny's three-day rescue from Mt. Everest in Nepal by Sir Edmund Hillary's personal pilot is a high point in his life. He suffered from both forms of the deadly acute mountain sickness. Donny had to summon all his will power to make his way down the mountain for two days to get to a lower altitude saving another climber on the way.

Lower altitude is the only cure. The climb down increasingly supplied more oxygen and stayed the death sentence. Helicopters and small planes can only fly at the lower altitudes Donny needed to reach in order to live. The helicopter almost crashed because of big rocks inside the clouds during the flight back to Katmandu. Weather also forced the helicopter down into a gorge for a night.

Donny has tried his hand as an adventure travel writer but has not met with much success. However, his bike trips across North and South America, Europe and most recently South Africa have led to many motorcycle travel articles that are enjoying some success.

Millions of viewers have enjoyed Donny and Tattoo Tony's technical segments on Canada's serial motorcycle show Biker TV as well as their appearances on The History of the Chopper.

Donny has no children in spite of three marriages and divorces. He takes onus for all three separations hesitating to call them failures. He views them as passing ships in life.

Paradoxically, Donny has taken up the sweet science of boxing at age 61, which he finds very fulfilling. He has great fun with the much younger boxers, some a third or of his age

that cannot figure out the old guy who hits back. He looks forward to going into the ring with younger boxers.

What is the future?

Donny will finish his twelve volumes of “Donny’s Unauthorized Technical Guide to Harley-Davidson 1936-Present.

He wants to revisit Everest.

He will ride his Harley’s.

He will remarry in a few years or maybe not.

Moreover, he is currently planning to take the time to walk the medieval pilgrim’s trek from France through northern

Spain.

Donny may relocate to the Pacific coast in Mexico to write, ride and drink some cold beer. Of course, the workaholic will need to start a business also to keep him occupied.

However, first he must deal with a 3-month Preliminary Hearing, similar to a Grand Jury to see if there is evidence to take him to trial for his membership position in his bike club. There is a court imposed publication ban on this hearing so nothing can be said at this point.

Hopefully by the time Volume III: The Evolution: 1984-1999 goes to print, Donny will be able to talk about this persecution; oops, sorry that was a Freudian slip for prosecution.